Short Term Water Heat Storage

An experimental and numerical investigation of phenomena that affect the degree of thermal stratification

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SHORT TERM WATER HEAT STORAGE

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PREFACE

The work in this dissertation has been carried out at the Division of Energy Engineering at Luleå University of Technology, Sweden. The financial support was given by Swedish National Board for Industrial and Technical Development and by Luleå University of Technology. Financial support was also received from Energiteknisk Centrum, ETC, in Piteå and from PiteEnergi AB.

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ABSTRACT

Thermal stratification was studied by measurements of the temperature and velocity fields in water heat storage's and by numerical simulations. The work includes experimental investigations in three model heat storage's of different size and comparisons to numerical simulations for two of the storages.

Thermal stratification is essential for most applications of water heat storage's. Different ways of measuring the degree of stratification were presented. Exergy efficiency was considered as best to use if the spatial resolution of the available temperature field is good. The thermal stratification is strongly dependent of the flow pattern in the storage. Mixing of hot and cold water near the inlet during charging can have a severe impact on the stratification. Much of the work was therefore focused on studies of this effect.

Velocity measurements were first performed with Laser Doppler Anemometry techniques and later with the use of a Particle Image Velocimetry technique based on double or multiple exposure photographs of particles in seeded water. These measurements focused on studies of the boundary layer at the wall. The results show that boundary layer velocities increase with the available height up to a limit, determined by the temperature difference, after which the dependence was very weak. They also show an exchange of water between the boundary layer and the core of the storage. A video based Particle Image Velocimetry technique was finally developed which opened the possibility to measure the in plane velocities in a cross section of a small model storage during complete experiments.

FLOW3D, a commercial software for flow simulations, was used to simulate measured velocity and temperature fields. Good agreement with the experimental results was found. This justified the use of FLOW3D to simulate the effect on stratification of parameters that could not be studied in the experiments.

The simulations for the small storage show that differences in the exergy efficiency obtained in the initial part of charging cycles, due to different charging conditions, tended to be evened out by heat diffusion during the continuation of the cycle. This was found to be true at least as long the Richardson number was kept high enough to avoid forced mixing.
An alternative choice of the length and velocity scales to use in the Richardson number was suggested. The width of the inlet slot was used as the characteristic length and the velocity of the inlet jet when it approaches the storage wall, as the characteristic velocity for the mixing.

With this definition for the Richardson number the simulations indicate that unstable charging can be expected for Richardson numbers below 0.5-0.05.

Further work is needed for verification of these findings for larger storage's. Still it is believed that the results of this work can be valuable for design optimization of short term water heat storage's.

**Keywords:** Water heat storage, thermal stratification, velocity fields, thermal mixing, PIV-measurements, flow simulations.
CONTENTS

DISSERTATION

INTRODUCTION S1

TEMPERATURE MEASUREMENTS S2

VELOCITY MEASUREMENTS S6
  Laser Doppler Anemometry S6
  Particle Image Velocimetry S9
  Video Based Particle Image Velocimetry S15

FLOW SIMULATIONS S18
  Theory S18
  The FLOW3D software package S19
  Simulations of the flow and mixing near the inlet S19

DISCUSSION AND CONCLUSIONS S27

RECOMMENDATIONS FOR FUTURE WORK S30

REFERENCES S31

Paper A: Formation and decay of the gradient zone in a thermally stratified hot water storage. A1-A101

Paper B: The establishing of a gradient zone and its effect on the velocity field in a water heat storage. B1-B13

Paper C: Temperatures and velocities near the gradient zone in a short term water heat storage. C1-C6

Paper D: Phenomena that affect the thermal stratification in water heat storage's. D1-D11

Paper E: Use of Video based Particle Image Velocimetry technique for studies of velocity fields in a water heat storage. E1-E36

Paper F: Thermal mixing near the inlet of a stratified water heat storage - Numerical simulations compared to experimental results. F1-F41
This dissertation consists of the following six papers

A. Dahl J, Hermansson R
   Formation and decay of the gradient zone in a thermally stratified hot water storage.

B. Dahl J, Hermansson R
   The establishing of a gradient zone and its effect on the velocity field in a water heat storage.

C. Dahl J, Hermansson R, Gren P and Benckert L
   Temperatures and velocities near the gradient zone in a short term water heat storage.

D. Dahl J, Hermansson R
   Phenomena that effect the thermal stratification in water heat storage's.

E. Dahl J, Hermansson R
   Use of video based Particle Image Velocimetry technique for studies of velocity fields in a water heat storage,
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   Thermal mixing near the inlet of a stratified water heat storage - Numerical simulations compared to experimental results.
   To be submitted for publication.
INTRODUCTION

Heat storage's are frequently used for example in district heating systems to even out variations in demand or production of heat. If surplus heat is available, the storage can be used to bridge the time wise difference between production and use of the heat. With a heat storage in the system it is possible to run boilers with optimum power output which will lead to higher average efficiency and less emission of pollutants. The storage can also reduce the need for installation of peak generation capacity.

Smaller heat storage's are used for instance in connection with heating system for individual buildings. This is commonly used with electric heating for utilization of cheap night tariffs, with wood fuelled furnaces for minimization of emissions, and with solar heating for elimination of the need for supplementary energy supply.

The most common storage medium and the easiest one to handle is water. Water heat storage's can be built for long term heat storing, for example in rock caverns, or as short term storage's often built as cylindrical steel vessels filled with water. In both cases it is essential that the temperature of the stored heat is maintained at a high level. This requires that the storage is thermally stratified.

Steel vessels can be built pressurised, filled to the top and connected to an expansion vessel, or having atmospheric pressure at the top in which case the storage can serve as expansion vessel for the whole network. In the case of atmospheric pressure the air is prevented from entering the storage by maintaining a steam cushion above the surface of the water. This requires the surface to be kept at a high temperature, close to 100°C. Otherwise the steam will condense on the water surface causing a decrease in steam pressure so that air will be sucked into the storage and cause corrosion damage to the steel vessel.

This is normally not a problem if the vessel is used for storing heat delivered from a boiler. The charging temperature of the water can then easily be kept at a high level. When the heat is delivered from sun collectors however, it is often not possible to maintain this high temperature level during charging. If the inlet is not properly designed the entering colder
water could then reach the water surface and cause the steam cushion to collapse.

Good knowledge about the thermal behaviour of the storage is essential both for proper design of storage's and for prediction of their behaviour when they are installed in a system. Such predictions are needed for valuation of the benefit of a storage in a system and for optimization of the operation of the system with a storage included.

The thermal behaviour of the storage in a network is determined by the temperature field inside the storage. This temperature field is to a large extent determined by the velocity field in the storage. A complete analytic solution to this problem, often transient, fully three dimensional and turbulent, is not possible to obtain. It is therefore necessary to rely heavily on experimental studies. Experiments are expensive however and it is often not realistic to make detailed measurements of all important phenomena. It was therefore decided first to increase the knowledge about the temperature and velocity fields in the storage by use of experimental methods, then to validate a numerical software that can be used in further investigations based on simulations and finally to use these simulations for generalization and extrapolation of the experimental results.

The ultimate goal for the project is to present a model which includes the most important phenomena that determine the storage's thermal behaviour and still is simple enough to use on small computers in engineering work.

The work made so far will be described in chronological order.

TEMPERATURE MEASUREMENTS

A laboratory model heat storage of size 1.2 m³ was constructed for the first experiments and equipped with a large number of temperature gauges. The storage was built as a steel vessel with diameter 800 mm and height 2400 mm. It was supplied with glass windows to enable visualisations and velocity measurements with LDA-technique.
A full description of the experiments performed in this storage and the results obtained is given in paper A. The aim of the investigations was to increase the knowledge about the phenomena that determine the degree of stratification in the storage by evaluation of experiments run with different charging conditions.

An important finding from these measurements is that the horizontal temperature distributions are fairly uniform below a level corresponding to about 10% of the volume for the conditions used in the tests.

There was a need for some aggregate measure of the result of an experiment, that is a way to measure the degree of stratification. This had earlier often been given as some kind of measure for the gradient zone, which is the volume between the hot and cold water in the storage. One possibility is to calculate the thickness of this zone. A narrow zone means good stratification. Another way is to calculate the temperature gradient in the zone, see for instance Sliwinski et al [17]. High values of the temperature gradient are desired. Comparisons can also be made on the basis of exergy content in the storage. This is probably the best way to value the maintenance of a high temperature level for the heat.

A straightforward calculation of these measures demands good spatial resolution of the temperature field, which in principle requires many temperature gauges in vertical direction. It is also possible to collect temperature data with small time intervals during charging or discharging of the storage. In this work the determination of these stratification measures was based on the time evolution of temperatures, which gives much better resolution.

Comparisons between different experiments clearly showed that low inlet velocities and high temperature difference between the entering water and the water in the storage gave better stratification regardless of which measure was used to quantify it.

It was found however that especially the thickness of the gradient zone can be misleading as a measure for the stratification. Cooling of the warm part of the storage can in fact decrease the thickness of the gradient zone although it should not be evaluated as an increase of the degree of stratification.
Defining a useful measure for the initial mixing of hot and cold water presented special problems. It would have been preferable to calculate the exergy efficiency during the charging cycle, but the poor spatial resolution of the measured temperatures close to the inlet where horizontal temperature distributions are not uniform did not make this possible. Instead the comparisons had to be based on the mixing volume and the initial penetration depth. These measures were calculated from the temperature evolution for gauges immediately below the level where the gradient zone first was established. They were found to be related to an inlet Richardson number, Ri, defined as

\[ \text{Ri} = \frac{g \cdot \beta \cdot \Delta T \cdot h}{v^2} \]

where
- \( g \) = acceleration of gravity
- \( \beta \) = coefficient of volumetric expansion
- \( \Delta T \) = temperature difference between incoming water and the water in the storage
- \( h \) = a length scale for the mixing
- \( v \) = a velocity scale for the mixing

This is a definition that for example is used by Cole [14] in his experimental study. He used the inlet velocity for the velocity scale and the distance between the inlet and the outlet for the length scale. This definition of the scales would mean that storage's, identical except that they have different height would give different Ri-numbers and thus indicate different mixing at the inlet. This is clearly not what could be expected and the Ri-numbers calculated with this definition of the length scale is only related to the storage used in his experiments.

The Richardson number can be interpreted as the ratio between work against buoyancy forces required for penetration a distance \( h \) into a region at a lower temperature and the kinetic energy of a water jet with a velocity \( v \). If \( h \) is the critical distance for penetration of hot water into cold water leading to significant mixing at the inlet and loss of stratification in this region, simple theoretical reasoning implies that the critical value for the Richardson number is 0.5.
In paper A the inlet velocity was used for the velocity scale and the unit length one meter for the length scale. This is no general definition and therefore not satisfactory and the results achieved are strictly speaking valid only for the storage used. However the qualitative dependence on the Ri-number should be correct. It was shown that there exists a critical value for the Ri-number which could be used to decide whether some charging conditions would give strong mixing or not.

For the storage and the definition of Ri-number used it was found that Ri-numbers greater than 0.25 gave little mixing and this mixing was only weakly dependent on the Ri-number. The same critical value for the Ri-number was found by Cole [14]. Lower values of the Ri-number gave strong mixing, which increased markedly with decreasing Ri-number. In order to solve the problem with the definition of scales in the Richardson number it was found necessary to measure velocities inside the storage.
VELOCITY MEASUREMENTS

Laser Doppler Anemometry

The first velocity measurements, which are described in paper A, were performed with the two component Laser Doppler Anemometry technique in the 1.2 m³ model storage described above. The LDA-equipment is shown in figure 1.

![EXPERIMENTAL SET UP LASER DOPPLER ANEMOMETRY](image)

Figure 1. The equipment for measurements with the Laser Doppler Anemometry technique

The laser that was used had an output effect that was too low to permit measurements in back scatter mode. Therefore the photo multiplier had to be placed on the opposite side of the storage and separately adjusted every time the measuring volume was changed. This made the measurements time consuming and made it difficult to capture the non stationary velocity field in the storage.

The technique was however suitable for measurements where the conditions were comparatively stable, for example in the boundary layer near the
wall. It was also possible to measure the time wise variation of velocities in one point in the storage in an easy way. However problems could arise with the measurements in the gradient zone where the varying refraction index could make it difficult to focus the photo multiplier on the measuring volume. The vertical and the tangential components of the velocities were measured but no turbulence quantities were recorded.

The effect of the temperature difference to the surrounding and the available height for the boundary layer on the maximum velocity in the boundary layer was investigated. It was found that the velocity increased with the available height for the boundary layer up to a value of this height of only a few decimeter. The maximum velocity did not increase noticeable when the height increased above this value. The suggested explanation for this was that the velocities increased with the available height as long as the flow was laminar. When transition to turbulence occurred the velocities stopped growing. Energy was then instead used to maintain the turbulence.

Figure 2 and 3 show the time evolution of the temperature and of the maximum velocity in the boundary layer at the same level (1.2 m from the top) in the storage. From these figures it is clear that the velocities are heavily effected by the gradient zone. This zone reaches the level for the measurements after about 80 minutes. The velocities start to decrease some minutes before (around 70 minutes) due to a decreasing height from the level for the measurements up to the lower part of the gradient zone. Before 70 minutes the velocities were very little effected by the available height, which was the distance to the top of the storage when the experiment started, here 1.2 m and then decreased continuously as the gradient zone moved downwards.

After the gradient zone had passed the level for the measurements the velocities reached a higher value due to higher temperature difference to the surroundings. When the temperature gradient was strongest at this level the velocities went down to zero.

It was also possible to measure the velocity profile in this boundary layer in experiments with stable conditions. The boundary layer thickness varied but was not greater than about 12 mm. The maximum velocity detected was less than 10 mm/s.
Figure 2. Temperature vs time at the level where the maximum velocities in boundary layer were measured, shown in figure 3.

Figure 3. Maximum velocity in the boundary layer vs time.
The velocities in the core of the storage were also measured and shown to be much more stable below the gradient zone than they were above this zone during charging experiments.

The initial mixing of hot and cold water near the inlet and the establishing of the gradient zone are also described in paper B prepared for presentation at the international conference North Sun in Borlänge Sweden, 1988. In this paper the relation between mixing volume and Ri-number and the effect on the velocity field from this gradient zone are discussed. Simple guidelines for the design of thermally stratified storage's are presented and the most important phenomena that affect the stratification are listed.

The overall behaviour of water heat storage's is determined by the temperature distribution inside the storage. However the temperature field is to a large extent determined by the velocity field since convective heat transport normally is more effective than pure diffusive transport.

There was still lack of information about the flow field in some important senses. First the field near the inlet was not possible to measure in the 1.2 m³ storage used due to the placement of the glass windows. Knowledge about this field is essential if the initial mixing at the inlets shall be described. Secondly it was not possible to measure the radial component of the velocities which determines the exchange of water between the boundary layer at the wall and the core of the storage. The weakness of the LDA-technique of only providing point information about velocities made it desirable to look for a method that could give field information.

**Particle Image Velocimetry**

At the Division of Experimental Mechanics, such measuring techniques were being developed. The first technique explored in this project was the White Light Speckle technique, where two flashes were used to obtain a double exposure photograph of particles in seeded water. The time delay between the two flashes had to be selected with consideration to the magnitude of the velocities to study.

The evaluation of these photographs was done by shining a laser beam through the negative or a positive contact copy of the film. The two spots of
the same particle will then give rise to interference fringes on a screen behind the negative which can be evaluated to give the direction and magnitude of velocities. These fringes will be stronger if there are many pairs of particle spots within the area of the shining laser beam. It was difficult to achieve a distinct light sheet and good illumination of the particles with this technique and the interference fringes were difficult to evaluate. These experiments are not described in the thesis.

Good illumination in a narrow light sheet is possible to achieve with a laser used as the light source and by spreading the laser beam into a light sheet with a cylindrical lens. The use of a mechanical shutter made it possible to achieve multiple exposures of the photograph which gives stronger interference fringes than double exposures. This technique, named Particle Image Velocimetry technique, PIV, was continuously developed at the Division of Experimental Mechanics. The interference patterns were read into a computer by a LCD-camera and a software was developed for automatic evaluation of the velocity field. A laser beam was arranged to shine through the film which was automatically moved in steps so that the contact copy of the film could be evaluated point by point.

In order to record and to get good illumination of a whole cross section of the storage it was necessary to build a smaller storage model with glass walls. The storage was equipped to enable experiments under well defined conditions and allowing both charging, discharging and stand still experiments. The experiments performed with PIV-technique in this storage model are reported in paper C and made in a small storage of rectangular shape. The storage had glass walls to enable visualisations and measurements with White Light Speckle and PIV-technique. The length of the storage was 400 mm, the width 250 mm and the height 420 mm.
Figure 4. Experimental set-up for the Particle Image Velocimetry technique based on multiple exposure photographs.

The experimental set-up is shown in figure 4. The measurements were performed in and near the boundary layer in a symmetry plane in the storage. The vertical velocity component and the component perpendicular to the wall were measured. Normally three or four exposures were used for evaluation of velocities. An example of a multiple exposure photo and the evaluated velocity field is shown in figure 5a and b. With a proper choice of the time delay between the exposures it was possible to achieve a very good resolution of the velocity field.
These experiments showed that there were large scale convection cells both below and above the gradient zone. Water was fed into the boundary later at the wall in the upper part of this boundary layer flow. Further down, water from the parts of the boundary layer closest to the core was flowing into the core of the storage. This exchange of water will certainly affect the stratification in the storage and has to be further studied.

Above the gradient zone, the main part of the boundary layer flow continued down to the gradient zone. Here the velocities were decreased by buoyancy forces and the flow was redirected towards the centre of the storage, causing a widening of the gradient zone.

At the level where the temperature gradient was strongest, the velocities became zero in this boundary layer. Below the gradient zone another convection cell was established with the same type of water exchange with the
core. The downward flow in the boundary layer causes an slow upward flow, relative to the mean flow, in the core of the storage.

Figure 6. Sketch of the principal flow in the storage a) during a stand still period and b) during charging of the storage.

Figure 6a shows a sketch of the principle flow field during a stand still period with a boundary layer at the wall caused by heat losses and figure 6b the field during a charging period. Vectors are not drawn in correct scales and the velocities in the core are much smaller than shown in the figure.

During a stand still period the boundary layer at the wall causes a slow upward movement in the core and water is fed to the boundary layer below the gradient zone and near the top. There is no boundary layer flow passing the gradient zone.

When the storage is charged, the mean velocity in the storage is added to the velocities shown above and causes an overall downwards movement in the core. The boundary layer above the gradient zone is now essentially fed with water from the inlet.
The effect from the gradient zone on the boundary layer flow that was
detected with this method was in agreement with the results from the LDA-
measurements. New knowledge was however gained about the exchange of
water between the boundary layer and the core of the storage. The PIV-
measurements indicated exchange of water between the boundary layer and
the core over the whole vertical extension of this layer, which was not
possible to conclude from the LDA-measurements. Large convection cells
were concluded to exist both above and below the gradient zone and there
was no flow across this zone.

The PIV-method used in this experiment was suitable for velocity measure­
ment were the velocity field was quite stable. The method has some weak­
nesses however:

- For fields with large variations in the magnitude of velocities it is
  necessary to take series of photos with different time delays between the
  exposures to cover the whole velocity range. The particles are not allowed
  to move more than 5-6 particle diameters between the exposures and must
  not move out of the area covered by the laser beam used for the evalu­
  ation.

- Many different time delays has to be used to cover wide velocity ranges. It
  is necessary to have some knowledge about this range to do proper choices
  of time delays. Otherwise the experiments must be repeated to catch the
  entire field. The time needed for the recording of these series of photo­
  graphs could mean that the velocity field has changed in transient experi­
  ments.

- The procedure for the evaluation is rather time consuming, for the devel­
  opment of the film, for making contact copies and for doing the computer
  evaluation of the field. Nevertheless figure 5 shows that quite nice veloc­
  ity fields can be achieved.

- There is an 180° uncertainty in the direction determined since it is not
  possible to know which one of the pictures of the same particle that is first
  recorded. The direction of the flow must be determined from visual obser­
  vations of the flow during the experiments.
Paper D is a summing up of the knowledge gained in the project until 1991, for presentation at the IEA Int. workshop on thermal energy storage and low energy buildings, in Germany 1991.

Many of the problems listed above could be avoided if many recordings of the particle movements could be made with small time delays, for example by use of video recording of the movements. This lead to the development of a video based PIV technique for velocity measurements.

**Video based Particle Image Velocimetry**

An ordinary video camera records the pictures at a rate of 25 frames per second and it is possible to make recordings during many hours. This makes it possible to document complete charging or discharging experiments for later evaluation of the velocity field.

It is afterwards possible to combine frames from the video tape with time delays from 1/25 of a second and in multiples of 1/25 s up to arbitrary time delays. It is also possible to use as many frames as desired and all this is determined after the experiments are performed. The evaluation can be performed with different time delays between frames to get most information out of the recordings, without running the experiment again.

The method allows measurements in one plane in the storage, continuously illuminated with a light sheet from a laser and a cylindrical lens. The water has to be seeded with suitable tracing particles as in the ordinary PIV-technique. A lot of information is gained by looking at the video recording of the flow and it is possible to do animation's to visualise flows that are to slow to see in real time.

There are no longer any problems with the direction ambiguity of the flow since it is known in which order the frames are captured and thus in which direction the particles are moving.

The principles of the method, estimates of its accuracy and the results that have been obtained with the method are described in paper E.
After the video recording the frames has to be captured from the video tape with adequate time delays between frames and fed into a computer. This is enabled by a frame grabber card in the computer and the free software IMAGE [12], which also is used for the image processing and determination of the co-ordinates for the particles on the different frames.

The next step is to determine which particle spots on the different frames that are spots of the same particle, recorded at different times. This is performed with a number of developed routines run in the commercial program IGOR [13]. These routines identifies the particle traces and determine the magnitude and direction of the velocities.

The accuracy of the method is primarily determined by the resolution of the video recordings and of the frame grabber card. Other sources of error are the calibration of distances in the viewed area, uneven illumination and the fact that particles might not follow the flow due to differing densities. The best measurements by a skilled user were estimated to give errors less than about 10% of the maximum detectable velocity, with the chosen time step. The maximum detectable velocity is limited since the present tracing routines does not allow movements larger than about one and a half millimetre during one time step.

The small storage was rebuilt before the experiments with this technique were performed. A storage of cylindrical shape was achieved by introducing a pipe made of Plexiglas with inner diameter 194 mm and height 420 mm in the centre of the rectangular storage, giving a model storage of 0.0123 m³. The volume between the Plexiglas pipe and the rectangular glass walls was filled with water, to have better optical conditions for the video recordings.

Figure 7 shows an example of a particle trace picture achieved by adding 10 frames captured at the rate of 1 frame/second from the video tape. Such pictures can be a powerful mean for visualisation of the flow field. Figure 8 shows the evaluated velocity field corresponding to figure 7. The field was detected near the bottom inlet of the storage during a discharging experiment.

The results from these measurements confirmed the results from the previous PIV-measurements concerning the exchange of water between the boundary layer at the wall and the core of the storage. It was also the first
Figure 7. Example of particle trace picture achieved with the video based Particle Image Velocimetry technique.

Figure 8. Velocity field near the bottom inlet, detected with the video based Particle Image Velocimetry technique during discharging of the storage.
measurements that showed the flow field near the inlets of the storage. The impact of different inlet velocities and different inlet temperatures in the flow and temperature fields were studied as shown in paper E.

Although the velocity field near the inlet could be documented in great detail, the knowledge about the temperature field was too sparse to make exact evaluations of the mixing of hot and cold water that occurs near the inlet. One of the goals for these experiments was therefore to generate information for the validation of the FLOW3D software for its further use in simulations of the thermal behaviour of water heat storage's.

FLOW SIMULATIONS

Theory

The flow field inside a water heat storage is never stationary, since heat losses and buoyancy forces are always present. In many cases, the flow is fully three dimensional and often turbulent. Simulations of this field requires the solution of the following set of equations

the continuity equation
\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{U}) = 0 \]

the momentum equation
\[ \frac{\partial \rho \mathbf{U}}{\partial t} + \nabla \cdot (\rho \mathbf{U} \otimes \mathbf{U}) = B + \nabla \cdot \sigma \]

where
\[ \sigma = -\rho \delta + \mu (\nabla \mathbf{U} + (\nabla \mathbf{U})^T) \]

and the energy equation
\[ \frac{\partial \rho H}{\partial t} + \nabla \cdot (\rho \mathbf{U} H) - \nabla \cdot (\lambda \nabla T) = \frac{\partial p}{\partial t} \]

Where H is the total enthalpy, given in terms of the static enthalpy h by
\[ H = h + \frac{1}{2} \mathbf{U}^2 \]

\( \rho \) is the fluid density, \( \mathbf{U}=(U,V,W) \) the fluid velocity, p the pressure T the temperature and t is time. Further B is a body force, \( \mu \) is the molecular viscosity and \( \lambda \) is the thermal conductivity.
To complete the set of equation for the seven unknowns, two equations are added, the equation of state
\[ p = \rho(T, p) \]
and the constitutive equation
\[ h = h(T, p) \]
For turbulent problems there are a number of transport equations for turbulent quantities added to the equations above. The number of equations is dependent on the complexity of the turbulence model adopted.

**The FLOW3D software package**

The simulations were performed with the commercial software FLOW3D Release 3.2 from AEA TECHNOLOGY, Harwell Laboratory UK [1]. Most of the simulations were run on a SUN Sparc IPX workstation but it was later possible to run the program on IBM RS6000, which decreased the time needed for a simulation by a factor of about 11. The basis of the code is a conservative finite-volume method.

The software permits fully three-dimensional calculations but in this case it was possible to use two-dimensional calculations due to the cylindrical symmetry of the storage. In this investigation, described in paper F, the flow near the inlet was simulated and compared to the results from the video based PIV-measurements described in paper E.

**Simulations of the flow and mixing near the inlet**

The comparisons were made for three experiments with the small 0.0123 m³ storage and one for the larger, 1.2 m³, storage. For the small storage, two charging and one discharging experiment was simulated. The flow conditions enabled use of laminar calculations. It was shown that the FLOW3D program predicted the velocity field in the storage in a good way, see figure 9 which shows the result of simulation of the measured field shown in figure 8.
Figure 9. Velocity field achieved in simulation of a discharging experiment. Compare to the measured field in figure 8.

Figure 10. Temperature evolution for five gauges in the storage as they are recorded in the experiment and predicted by FLOW3D.
The temperature evolution for gauges in the storage was also predicted with the program with good agreement to the measured evolution as shown in figure 10. It was concluded that the software should be possible to use for simulations of the mixing that occurs near the inlet of the small storage.

However when the simulations were extended to cover the experiments performed in the big storage some problems arose that are discussed in APPENDIX A to paper F. If the same number of grid points was used in this bigger storage there was problem with the mass continuity evaluated from the velocities predicted in the centre of cells. There were small grid induced velocities that caused mixing in the gradient zone. The temperature evolution in the storage was therefore not predicted correctly, giving less steep gradients in the simulation than found experimentally.

The only way found up till now to master this problem was to use grids that were as fine as those used in the small storage. In this way, by use of 10000 computation cells, the temperature evolution in the larger storage was well predicted but with very long computation times. It was concluded that the software is not suitable to use for simulations in full scale storage's, with reasonable computation time using the computers available for this study.

With the present status of the program and the available computers, the simulations has to be performed for small storage's and the results extrapolated to large storage's by scale analysis.

The numerical study on parameters that affect the initial mixing during a charging cycle is described in APPENDIX B to paper F.

The comparisons between different simulations in this investigation were based on an exergy efficiency defined as the actual exergy content in the storage divided by the highest possible content if no mixing had occurred, that is if all heat in the storage would have inlet temperature.

The effects of inlet temperature, inlet velocity, inlet mass flow, temperature difference, inlet diameter, width of the inlet slot and the diameter of the storage on the evolution of the exergy efficiency during charging was studied by the simulations.
It was found that for stable charging of this small storage there were significant differences between the different simulations in the initial phase of the charging process. In most of the simulations these differences were reduced during the continued charging and most of the simulations showed the same exergy efficiency after charging the storage to a bit less than 25%. This was concluded to be an effect of the heat diffusion from hot to cold water which tended to compensate for the differences that were achieved in the initial phase, see figure 11.

There were however two simulations for stable charging that gave smaller exergy efficiencies than the other. One was simulating a storage charged with very low flow from an inlet with small diameter. The other simulated a storage with large storage diameter but the same inlet diameter and flow as for the reference case. Both these cases causes the first incoming jet to be spread over a large horizontal area when it enters the storage. This will cause more mixing in the contact between hot and cold water and strong heat diffusion since the temperature gradient becomes large in this contact region. There is also a larger distance up to the wall in these cases and there will be more mixing due to the widening of the jet and due to reintrainment of colder water into the jet. These differences will also be evened by heat diffusion but only if the heat is stored for a long time in the storage.

As described above, there had been problems to define the correct scales to use in the Richardson number for predictions on whether certain charging conditions would cause severe mixing or not. It had earlier been concluded that the Ri-number is suitable for such predictions but the scales that were used had been valid only for the storage's used in the experiments. New scales for velocity and length were proposed for use in the calculations of the Richardson number to be used for assessment of the stability of the charging. It is suggested that the width of the inlet slot is used as the length scale and the velocity of the inlet jet as it approaches the wall as the velocity scale.

When the Ri-numbers are calculated for the stable experiments discussed above it results in Ri-numbers between 1 and 288 with the new scales and between 6 and 15 000 with the old ones. That is, both ways to calculate the Ri-number indicate stable charging if the critical value is 0.25. The simulated transient temperature field for stable charging can be illustrated by
figure 11 which shows the almost perfect stratification obtained already after charging to about 3%.

The pattern is quite different for the two simulations with lower Ri-numbers. With the new definition, the Ri-numbers for these cases are 0.003 and 0.05 compared to 0.13 and 2.4 with the definition suggested in [14]. Figure 12 shows the temperature pattern after charging to 25% for the lowest Ri-number and 13 that for the next lowest. The dramatic penetration of the hot water into the cold part of the storage is evident for the lowest Ri-number. For the next lowest Ri-number, the jet penetrates into the cold water but the buoyancy forces stabilise the charging later in the transient. At 25% charging, figure 13, the storage is reasonably stratified. Apparently, the critical Ri-number is above 0.05 with the new definition of the scales but the practical importance of the instability obtained at this Ri-number is not large since the exergy efficiency recovers to the same level as for stable charging.

Recalculation of the Richardson numbers presented in paper A using the new definition for length and velocity scales indicates a drastic increase in the mixed volume for Ri below about 0.015.
Figure 11. Predicted temperature field after 3% charging of the storage with Richardson number 40.
Figure 12. Predicted temperature field after 25% charging of the storage with Richardson number 0.003.
Figure 13. Predicted temperature field after 25% charging of the storage with Richardson number 0.05.
DISCUSSION AND CONCLUSIONS

Different techniques for the evaluation of the temperature and velocity field in stratified water heat storage's were applied in this project.

The temperature measurements showed that the temperatures were quite fast evened in horizontal direction. The mixing of hot and cold water near the inlet of the storage was shown to be related to a modified Richardson number, the quotient between buoyancy and inertia for the incoming water. The correct scales to use in this number were not known and motivated further investigation.

Velocity measurement with Laser Doppler Anemometry showed that the velocities in the boundary layer at the wall increased with the available height up to a value of this height of a few dm. For larger values of this height the velocities were mainly determined by the temperature difference to the surrounding.

The first measurements of the radial velocities in the storage were performed with the Particle Image Velocimetry technique. These measurements showed that there were large scale convection cells above and below the gradient zone, which were responsible for the exchange of water between the boundary layer and the core of the storage. Water was fed into the boundary layer at the top of this layer and a small outflow was indicated over the rest of its vertical extension.

A new video based PIV-technique was developed in the project and proved to be useful for measurements of small velocities. A lot of effort was put into the tests of this technique and the results were very encouraging despite some encountered problems of resolution and accuracy.

The technique made it possible to measure the velocity field in the whole storage and to record complete experiments for later evaluation.

This information about the velocity field and the simultaneous temperature measurements made it possible to validate the software FLOW3D against experiments. The software package proved to give good agreement between simulated and experimental velocity and temperature fields in the
small 0.0123 m³ storage and also between the simulated and experimental temperature field in the larger 1.2 m³ storage.

Simulations for the larger storage however demanded the same grid resolution as in the small storage to give good results. The number of grid points was therefore increased roughly with the square of the geometry scale for two-dimensional simulations, if the predictions should give good agreement with the experimental results.

If simulations for stratified storage's are performed with this software careful control of the mass continuity calculated with velocities at cell centres is necessary.

The good agreement found between simulations and experimental data encouraged the use of the software for simulations of the impact from different variables on the mixing that occurs near the inlet during charging of a water heat storage.

The simulations indicate that heat diffusion has a greater impact on the thermal behaviour during the initial phase of the charging than was suggested in earlier work. This is probably to a large extent an effect of the scale for the simulations. In larger storage's with wider inlets the exergy losses due to heat diffusion across the horizontal contact area will represent a much smaller fraction of the incoming exergy flow. More work is needed for the scaling of this effect to larger storage's or for simulation of this effect in a larger scale.

The simulations show that as long as the charging conditions are stable, as judged by the Ri-number, there are no large differences between the separate charging conditions after 10-25% charging of the storage. The initial differences disappear to a large extent during the continued charging.

The effect should be the same in larger storage’s even if the process is slower due to the wider gradient zone. The temperature difference between the hot and the cold water is normally higher and the residence time for the water in the storage is longer in a full size storage. The conclusion would thus be that if the whole storage cycle is considered the heat diffusion will eliminate much of the initial differences achieved if the charging conditions are stable.
Even if the simulations show that unstable charging conditions is the most important reason for a low exergy efficiency, it is still important to consider other factors which tend to reduce the exergy efficiency. Results of the simulations indicate that the size of the inlet slot would not be too large and the inlet diameter not too small. In both cases more contact between hot and cold water during the initial phase is obtained.

The simulations indicate that the Richardson number is not suitable for predictions of the exergy efficiency for stable chargings, since it does not account for all important variables. This is a conclusion that should be valid also for larger storage's.

The new scales proposed for the length and the velocity in the Ri-number are more related to the geometry where the mixing occurs and should enable more precise predictions of the risk for undesirable mixing of hot and cold water during the initial phase of the charging. These scales should be suitable to use also for larger storage's.

For the small cylindrical storage, the critical value for the Ri-number, calculated with the new scales, is above 0.05. At this Ri-number the hot water does not penetrate far into the cold water, however. As charging progresses the storage becomes stratified and the exergy efficiency reaches about the same levels as for stable charging. For Ri-numbers that are an order of magnitude lower, the mixing becomes severe and the exergy efficiencies much lower.

In larger storage's the incoming jet will be turbulent and more work is needed for determination of the critical Ri-number. It is necessary to evaluate the influence from the widening of the jet and from reintrainment for the turbulent case. The main flow towards the wall is expected to be roughly the same however.

Taking into consideration the weak influence of the Ri-number on the exergy efficiency when some penetration of hot water occurs at the wall, it is expected that a storage designed for a Ri-number of 0.05-0.5 will give reasonable exergy efficiencies also for the turbulent case. Additional work is needed to verify this.
RECOMMENDATIONS FOR FUTURE WORK

The remaining uncertainty about the value of the critical value for the Ri-number in large storage's is not satisfactory and should be addressed in future work. More simulations and experiments are needed to establish this limit.

One phenomenon that has not been addressed properly in the simulations made so far is the impact of the boundary layer flow on the stratification in the storage. It seems promising to use the present software for simulations for the small storage where conditions are laminar. However, for large storage's also the boundary layer flow can be turbulent. The turbulence models must therefore be tested and validated against experiments in a large storage. The effect from the boundary layer flow can be significant for long residence times in the storage.

The video based Particle Image Velocimetry technique showed a good potential for further development. For example should it be possible to develop the particle tracing routines to give more and maybe also more accurate information from the video recordings. With this technique it could be possible to record the velocity field also in a large storage where the conditions can be turbulent.

Future research must also be directed towards the prediction of the thermal behaviour of full scale storage's.

One way to do this would be by use of the results achieved on small storage's in the project to formulate a simplified model for large storage's, which then can be validated against the performance of some existing full scale storage.

Another way would be to try to extend the simulations to cover larger storage's. This however requires that the present problems regarding the mass continuity calculated at cell centres are solved. The simulations performed show that the problems are possible to solve by grid refinement which leads to long computation times. Since the software is continuously developed it is possible that some solution to the problem might be implemented. It might be worth looking at other software's to investigate whether they give the same problems.
Other storage geometries should be studied, for example rock caverns where the natural convection could be expected to have major influence on the stratification due to long residence times.

REFERENCES


FORMATION AND DECAY OF THE GRADIENT ZONE IN A THERMALLY STRATIFIED HOT WATER STORAGE

Jan Dahl
Roger Hermansson

Division of Energy Engineering
Luleå University of Technology
ABSTRACT

Thermal stratification is necessary to get the best performance for a water heat storage. The initial formation and further development of the gradient zone in a short term storage has been studied.

Temperatures and velocities in a 1.2 m$^3$ storage are measured. The initial part of a charging period is shown to have a major influence on the gradient zone and the initial penetration depth is strongly related to the Richardson-number. For Ri-numbers greater than 0.2 the initial penetration depth is almost independent of Ri and for smaller values it is shown to be proportional to $1/\sqrt{\text{Ri}}$.

The velocities in the boundary layer at the wall are shown to be roughly proportional to the temperature difference between the water and the surroundings and only to a small extent dependent on the available height. The flow in the boundary layer is laminar, turbulent or in transition to turbulence and measured velocities are of the magnitudes $10^{-3}$ to $10^{-2}$ m/s.

Benard convection, induced by the heat losses at the top of the storage, has a strong influence on the temperatures above the gradient zone during stand still periods.

Temperatures are shown to be equal in horizontal direction, at least within the accuracy of the used temperature gauges.

KEYWORDS

Water heat storage, thermal stratification, gradient zone, velocity measurements, temperature measurements, penetration depth.
# CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>2. PHYSICAL BACKGROUND</td>
<td>4</td>
</tr>
<tr>
<td>3. THEORY</td>
<td>8</td>
</tr>
<tr>
<td>3.1 General background</td>
<td>8</td>
</tr>
<tr>
<td>3.2 Non-dimensional analyses</td>
<td>9</td>
</tr>
<tr>
<td>3.3 Heat conduction</td>
<td>14</td>
</tr>
<tr>
<td>3.4 The stratified water heat storage</td>
<td>14</td>
</tr>
<tr>
<td>4. STORAGE EFFICIENCY</td>
<td>18</td>
</tr>
<tr>
<td>5. BACKGROUND</td>
<td>21</td>
</tr>
<tr>
<td>6. EXPERIMENTAL SETUP</td>
<td>26</td>
</tr>
<tr>
<td>7. EXPERIMENTAL RESULTS AND DISCUSSION</td>
<td>34</td>
</tr>
<tr>
<td>7.1 Qualitative conclusions from temperature measurements</td>
<td>35</td>
</tr>
<tr>
<td>7.2 Quantitative treatment of measured temperatures</td>
<td>46</td>
</tr>
<tr>
<td>7.2.1 Temperature gradient</td>
<td>46</td>
</tr>
<tr>
<td>7.2.2 Thickness of the gradient zone</td>
<td>50</td>
</tr>
<tr>
<td>7.2.3 Initially mixed volume, initial penetration depth</td>
<td>53</td>
</tr>
<tr>
<td>7.2.4 Richardson number</td>
<td>55</td>
</tr>
<tr>
<td>7.2.5 Energy and exergy efficiency</td>
<td>60</td>
</tr>
<tr>
<td>7.3 Evaluation of the velocity measurements</td>
<td>68</td>
</tr>
<tr>
<td>7.3.1 Measurements in the boundary layer</td>
<td>69</td>
</tr>
<tr>
<td>7.3.2 Measurements in the core</td>
<td>82</td>
</tr>
<tr>
<td>8. CONCLUSIONS</td>
<td>91</td>
</tr>
<tr>
<td>ACKNOWLEDGEMENTS</td>
<td>97</td>
</tr>
<tr>
<td>REFERENCES</td>
<td>98</td>
</tr>
</tbody>
</table>
1. INTRODUCTION

The basic task of heat storage is to bridge the time-vise or local gap between heat demand and supply. If the mismatch between supply and demand is caused by changes on either side, the differences between the heat production and heat demand may be solved by means of a heat storage.

First of all a definition of the headline "Short term water heat storage" is required. Short term means that the heat storage is designed for the dominant periodicity of either the load or the supply, normally a day or a weekend. Water heat storage is the storage of sensible heat in saturated water. This also includes a liquid water storage with a steam cushion on the top.

Short term storage systems are applied in the industry, in nuclear- and other power plants for feed water heat storage, in the paper and pulp industry in order to operate the bark fired boilers, or in the backpressure system for optimizing the production of steam and electricity.

Heat storage in a district heating system is standard in many countries. In a heating plant a storage makes it possible to run the boiler with a higher efficiency and with less pollution to the environment. The heat storage can bridge sudden peak demands of heat instead of starting up the reserve unit. A heat reserve is available when there is a breakdown in a production unit. A water reserve is available if there is a leakage in the piping system.

If one want to store waste heat from different types of process industries for use in a district heating system there are some special problems. There are strong fluctuations in the heat supply depending on the production at each industry. This can result in many charging and discharging cycles per every 24 hours. Every time the gradient zone will pass through the inlet and outlet of the storage there will be a growth of the gradient zone. After a few cycles the broadening is so great that you must discharge the storage completely and start up with the building of a new gradient zone.

In a combined heat and power plant the production of electricity is dependent on the heat demand. During peakloads in the power production the storage can be used as a heat sink.
Space heating and domestic hot water systems are well suited for short term water heat storages as both supply and load are irregular. If the heating system is based on electricity, the heat can be produced with cheap electricity during the night or during low load periods in the day. In a family house with a wood fired space heating system one can, during a few hours in the evening, charge such a big storage that the heat is sufficient for the next day.

In solar thermal systems where the supply is uncontrolled, a storage system is needed, if the instantaneous load is lower than the maximum thermal production rate.

To get the best performance for a water heat storage it is necessary to maintain a good thermal stratification (increasing temperature with height). The main purpose is to conserve the quality or exergy of heat stored.

If one want to make a detailed study of the storage efficiency in different situations it is necessary to study one complete cycle (charging and discharging of one storage volume) and analyse it carefully in the energy and exergy point of view.

However from some simple examples it is possible to get an understanding of a few important facts.

Assume a heat storage that is charged up to 50 % of the total energy capacity. Compare three different thermal situations.

a) $T_{\text{max}}$ in the upper part of the storage, an infinite temperature gradient and $T_{\text{min}}$ in the lower part of the storage.

b) $T_{\text{max}}$ in the top, a linear temperature gradient to the bottom and $T_{\text{min}}$ in the bottom of the storage.

c) A fully mixed storage with $T_{\text{med}} = (T_{\text{max}} + T_{\text{min}})/2$ in the whole storage.

The assumption above gives the result that the energy content is equal. However the value of the temperature level is not always equal. Assume that only temperatures over $T_{\text{med}}$ are useful in the external system.
That gives

a) 100 % of the charged energy has a useful temperature.
b) 75 % of the charged energy has a useful temperature.
c) 0 % of the charged energy has a useful temperature.

The importance of a good temperature stratification becomes quite clear from this example.

A use of the definition of exergy shows that the temperature level and the stratification in the storage is very important as the value (the Carnot efficiency) of the exergy is increasing with the temperature. The best storage due to the exergy content is the one with infinite temperature gradient, and the worst one is the fully mixed storage with zero gradient.

As a conclusion one can see that it is important to maintain the stratification without any mixing in order to get the maximum temperature gradient and in that way keep the quality of the heat in the storage.

To get a better understanding of the phenomena that disturbs the temperature stratification and to be able to analyse the complex thermal situation, comprehensive experimental studies involving temperature and velocity measurements are needed.
2. PHYSICAL BACKGROUND

Physical properties like for example specific heat, density, kinematic viscosity, heat diffusivity and heat conductivity are very important for the medium in a heat storage.

Water has many good qualities in this sense.

The specific heat times density gives a good indication of the quality for a medium. Table 2.1 shows that the volumetric capacity for water is better than for other storage media.

<table>
<thead>
<tr>
<th></th>
<th>Lower operation temp (°C)</th>
<th>Upper operation temp (°C)</th>
<th>Density (kg/m³)</th>
<th>Specific heat capacity (kJ/kg.K)</th>
<th>Volumetric heat capacity (kWh/m³.K)</th>
<th>Volumetric energy density (kWh/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water 1 bar</td>
<td>0</td>
<td>100</td>
<td>1000</td>
<td>4.19</td>
<td>1.16</td>
<td>116</td>
</tr>
<tr>
<td>10 bar</td>
<td>0</td>
<td>180</td>
<td>1000</td>
<td>4.19</td>
<td>1.03</td>
<td>185</td>
</tr>
<tr>
<td>Water and Ethylen glycol</td>
<td>6</td>
<td>100</td>
<td>1075</td>
<td>3.62</td>
<td>1.00</td>
<td>100</td>
</tr>
<tr>
<td>Heat Transfer Oil</td>
<td>20</td>
<td>250</td>
<td>750</td>
<td>2.50</td>
<td>0.52</td>
<td>120</td>
</tr>
<tr>
<td>Hot Rocks</td>
<td>20</td>
<td>100</td>
<td>2700</td>
<td>0.80</td>
<td>0.60</td>
<td>60</td>
</tr>
<tr>
<td>Iron</td>
<td>20</td>
<td>350</td>
<td>7820</td>
<td>0.46</td>
<td>1.00</td>
<td>350</td>
</tr>
</tbody>
</table>

Table 2.1 Volumetric capacity and volumetric energy density for different heat storage media

It is important to keep in mind that the available temperature level is different for different media. If you use the volumetric energy density you must define the maximum and minimum temperature level in each case.

An unpressurized water storage has of course a maximum temperature of 100°C. The maximum available temperature difference is depending on the specific technical situation and in practice not greater than 70°C.

Heat transfer oil with a boiling point around 300°C and a max. temperature of about 250°C becomes more interesting when you are looking at the volumetric energy density table 2.1.
The viscosity for oil is rather high and this gives a more stable temperature stratification and less mixing in the storage.

However there are some disadvantages with heat transfer oil. The high temperature level will demand more insulation, the price for heat transfer oil is quite high and there is a need of one more heat exchanger, at least in a Swedish heating system.

The low price and the high volumetric energy density for water implies that water is a very useful medias in heat storage systems.

As we have stated before, a good working water heat storage has theoretically an infinite temperature gradient and no mixing.

The density variation due to the variations in temperature gives a natural and stable stratification with hot water in the top and cold water in the bottom of the storage and no variation in horizontal direction, fig. 2.1.

![Graph showing density vs temperature](image)

**Figure 2.1** Density for water vs temperature.

Heat losses at the bottom also give a stable temperature stratification.
However several phenomena can disturb the thermal stratification.

The heat losses to the surroundings give rise to natural convection near the walls. The buoyancy force is the cause of the motion and characterize the flow. Important parameters are, temperature difference between the fluid and the wall, characteristic length (available height) of the storage, volumetric expansion and kinematic viscosity for the water and the temperature level in the surroundings.

![Kinematic viscosity for water vs temperature.](image)

Figure 2.2   Kinematic viscosity for water vs temperature.

Near the inlet, forced convection due to the inlet velocity give rise to a turbulent situation with mixing between hot and cold water. Inertia forces and buoyancy forces counteracts. When the gradient zone is a bit away from the inlet, viscous forces counteracts the inertia forces and therefore the kinematic viscosity due to the viscous dissipation becomes more important.

Heat losses at the top cause an unstable temperature stratification and give rise to Benard convection, which is important at least during stand still periods.

The heat diffusivity for water is also important during stand still periods.
Heat conduction in the walls is around 100 times that of water. The heat capacity for the walls becomes important at least for small storages and this will decrease the stratification.
3. THEORY

3.1 General background

A free convection flow is one produced by buoyancy forces. Temperature differences are introduced and the consequent density differences induce the motion.

When temperature differences are introduced but they are not large enough to modify the flow one have a forced convection flow.

The temperature variations in a convective flow give rise to variations in the properties of the fluid, in density and viscosity for example.

An analysis including all these effects is so complicated that some approximation becomes essential.

In the Boussinesq approximation the variation of all other fluid properties than the density are completely ignored and variations of the density are ignored except insofar they give rise to a buoyancy force.

Thus the basic equations in the Boussinesq approximation form will be

\[ \nabla \bar{u} = 0 \]  \hspace{1cm} (3.1)

\[ \frac{D\bar{u}}{Dt} = (-1/\rho) \nabla p + \nu \nabla^2 \bar{u} - g\beta \Delta T \]  \hspace{1cm} (3.2)

\[ \frac{D T}{Dt} = k \nabla^2 T \]  \hspace{1cm} (3.3)

(3.1) is the continuity equation in constant density form, where \( \bar{u} \) is the velocity vector

(3.2) is Newton's second law of motion for a fluid of constant density (Navier Stokes eq.). The term \( g\beta \Delta T \) represents the buoyancy force, where \( \beta \) is the coefficient of volumetric expansion and \( T \) is the absolute temperature, \( p \) is the pressure in the fluid, \( \nu \) is the kinematic viscosity, and \( D/Dt \) is the material derivative.
where

\[ \frac{D}{Dt} = \frac{\partial}{\partial t} + u \frac{\partial}{\partial x} + v \frac{\partial}{\partial y} + w \frac{\partial}{\partial z} \]

(3.3) is the energy equation where

\[ \frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = \frac{\partial T}{\partial t} + u \nabla T \]

and \( u \nabla T \) is the convection term representing the transport of heat by motion. \( \kappa \nabla^2 T \) is the heat conduction term where the thermal diffusivity \( \kappa \) is

\[ \kappa = \frac{k}{\rho c_p} \quad (3.4) \]

\( \rho \) is the density, \( c_p \) is the specific heat and \( k \) is thermal conductivity.

Eq. 3.2 and 3.3 must now be considered simultaneously, as both involve both \( u \) and \( T \).

In the free convection problem both \( u \) and \( T \) depend on each other (the velocity distribution is governed by the temperature distribution but the temperature distribution depends through the convection of heat on the velocity distribution) so there is no possibility to solve one equation independently of the other, as sometimes could be done in forced convection problems.

As we have nonlinear partial differential equations in \( u \) and \( T \) there are essential mathematical problems in solving the equations.

### 3.2 Non-dimensional analyses

**Forced convection**

If the velocity field is known and uneffected by the temperature field the temperatures can be determined by the energy equation (3.3).
Thermal similarity between two systems with steady convection exists when the Pe-numbers are equal.

where \( Pe = RePr = uL/\kappa \) \hspace{1cm} (3.5)

and \( Re = uL/\nu \) and \( Pr = \nu/\kappa \) \hspace{1cm} (3.6)

\( L \) is a characteristic length for the problem.

Full similarity requires equality of both the Re-number and the Pe-number.

\( Pr = 1 \) gives identical temperature and velocity distributions in a laminar boundary layer

The heat transfer quantity is of great interest and a nondimensional form of this is

\[ Nu = hL/k \] \hspace{1cm} (3.7)

where \( h \) is the heat transfer coefficient per unit area

Dimensional considerations indicate that

\[ Nu = f(Re,Pr) \] \hspace{1cm} (3.8)

Neither \( Re \) nor \( Pr \) involves the temperature, which means that also \( h \) is independent of the temperature.

Natural convection

Non dimensional analyses of the continuity eq. (3.1), and the energy eq. (3.3) gives that dynamical similarity depends on the Gr-number and the Pr-number.

In this situation the Gr-number indicates the type of flow to be expected, whether the flow is laminar or turbulent.
In natural convection the buoyancy force is the cause of the motion so it must always be greater than each one of the inertia and viscous forces.

Assume that the inertia force is of the same order of magnitude as the buoyancy force or

\[ |\mathbf{u}\mathbf{u}| \sim |\mathbf{g}\beta \Delta T| \]  \hspace{1cm} (steady convection) \hspace{1cm} (3.9)

that is

\[ \frac{u^2}{L} \sim g\beta \Delta T \]  \hspace{1cm} (3.10)

and

\[ u \sim (g\beta \Delta TL)^{1/2} \]  \hspace{1cm} (3.11)

which make it possible to predict the velocity scale, that is how fast the fluid will move as a result of the temperature difference.

If one now compare the orders of magnitude of the inertia and viscous forces one get

\[ \frac{|\mathbf{u}\mathbf{u}|}{|\nabla \mathbf{u}|} \sim \frac{uL}{v} \sim (g\beta \Delta TL^3/v^2)^{1/2} = \text{Gr}^{1/2} \]  \hspace{1cm} (3.12)

This tells us that, when the Gr-number is large the viscous force is negligible compared to the buoyancy and inertia forces.

To deal with the case that Gr is small you must start with the alternative assumption that the viscous force is comparable to the buoyancy force.

\[ |\nabla \mathbf{u}| \sim |g\beta \Delta T| \]  \hspace{1cm} (3.13)

which gives

\[ u \sim g\beta \Delta TL^2/v \]  \hspace{1cm} (3.14)

and

\[ \frac{|\mathbf{u}\mathbf{u}|}{|\nabla \mathbf{u}|} \sim \text{Gr} \]  \hspace{1cm} (3.15)
which indicates that small Gr-number implies negligible inertia forces, compared to buoyancy and viscous forces.

Consequently

large Gr-numbers implies large inertia forces
small Gr-numbers implies small inertia forces

In convection problems it is important to determine which processes are important for the temperature distribution.

Compare the orders of magnitude in the same way as before for the convection term and the diffusion term in eq. (3.3)

\[ \frac{|uVT|}{|kV^2T|} \sim \frac{uL}{k} \]

(3.16)

Large Gr-numbers

\[ u \sim (g\beta \Delta TL)^{1/2} \]

(3.11)

That gives

\[ \frac{|uVT|}{|kV^2T|} \sim Gr^{1/2}Pr \]

(3.17)

Small Gr-numbers

\[ u \sim g\beta \Delta TL^2/v \]

(3.14)

That gives

\[ \frac{|uVT|}{|kV^2T|} \sim GrPr \]

(3.18)

Where \( uL/k \) and \( uL/v \) is the Pe-number and Re-number both dependent on \( u \), in a similar way as GrPr.

\[ Re = f(Gr, Pr) \]

(3.19)

and for the heat transfer situation dimensional analysis gives a similar relation as for forced convection
\[ \text{Nu} = f(\text{Gr}, \text{Pr}) \]  
\hspace{1cm} (3.20)

For Pr-numbers around 1 and for large Gr-number the convection always dominates over conduction, in the same way as inertia forces over viscous forces.

However this is based on the assumption that the length scale is L. This is not valid in the boundary layer where the length scale is \( \delta \ll L \).

The heat conduction is always responsible for the temperature distribution in the fluid near the wall.

The correct inference is that when \( \text{Gr}^{1/2}\text{Pr} \) and \( \text{Gr}\text{Pr} \) are large the flow near the wall will have a boundary layer character.

The boundary layer can either be turbulent or laminar.

Over a vertical heated plate laminar flow is observed in the lower part. In this region good agreement is obtained between theory and experimental investigations.

An often used assumption for the velocity profile is

\[ \frac{u}{u_x} = (y/\delta)(1-y/\delta)^2 \]  
\hspace{1cm} (3.21)

which gives the boundary layer thickness

\[ \delta/x = 3.93\text{Pr}^{-1/2}(0.952 + \text{Pr})^{1/4}\text{Gr}_x^{-1/4} \]  
\hspace{1cm} (3.22)

Further up the plate the laminar flow becomes unstable and undergoes an transition which results in a fully turbulent boundary layer.

A calculation of the GrPr-number indicates where the transition occurs.

For a fluid with \( \text{Pr} = 1 \) laminar flow can be observed up to a Gr-number around \( 10^9 \), even though it becomes unstable at a Gr-number around \( 10^5-10^6 \).

For \( \text{GrPr} > 10^9 \) turbulent boundary layer  
\hspace{1cm} (3.23)
In the turbulent free convection case a normally used expression is

\[
u/u_0 = \eta^{1/7} \frac{1}{(1-\eta)^4} \eta = y/\delta
\]  

(3.24)

One way to determine whether one have natural or forced convection is through a calculation of the Gr- and Re-numbers.

for \( Gr/Re^2 \gg 1 \) natural convection  

(3.25)

and for \( Gr/Re^2 \ll 1 \) forced convection  

(3.26)

3.3 Heat conduction

The general heat conduction equation can be achieved from an energy balance over an elemental volume using the energy equation (3.3) without the convective term

\[
\frac{\partial T}{\partial t} + q = \kappa \nabla^2 T
\]  

(3.27)

where \( q \) is the heat transfer rate.

The one dimensional steady state formulation is called the Fourier's law

\[
q = -kA \frac{dT}{dx}
\]  

(3.28)

3.4 The stratified water heat storage

The thermal situation in a stratified water heat storage is much more complicated than most of the problems treated in the litterature.

The complexity in every situation is of that order that a complete theory is impossible to present.

The knowledge about temperature and velocity field must be based on a comprehensive experimental work.
However a nondimensional analysis of certain situations gives qualitative information and is a good help when you are trying to analyse different phenomena.

During **charging**, the most important time phase is in the beginning of the charging period, when the temperature gradient is established in top of the storage.

Near the inlet there is **mixing** between hot incoming water and cold water in the storage.

Buoyancy forces are acting against inertia forces. In the beginning the length scale for the mixing is decreased, depending on the large temperature difference.

After a while it is reduced when the temperature is increasing in the top of the storage, but the inertia forces are constant. This will cause a mixing further down in the storage.

The quotient between buoyancy forces and inertia forces

\[
|g\beta \Delta T|/|u\nabla u| \text{ can be written as } \tag{3.29}
\]

\[
g\beta \Delta TL/u^2 = Ri \tag{3.30}
\]

which is known as the modified **Richardsson-number**.

After some time the temperature gradient is established under the mixing zone.

Now there is a situation where inertia forces and viscous forces counteracts. The quotient between inertia and viscous forces

\[
|u\nabla u|/|v\nabla^2 u| \text{ can be written as } \tag{3.31}
\]

\[
uL/v \tag{3.6}
\]

which can be identified as the **Re-number**.
During the charging period the thermocline is moving downwards and the distance to the inlet is increasing. The kinetic energy of the incoming water is now dissipated due to the viscosity in the water.

A nondimensional analysis is not meaningful in this region because we have not enough information about the turbulent length scales. Numerical calculations with some kind of turbulence model could be fruitful. Nevertheless a knowledge about the turbulent kinetic energy and the length scales is also in this case necessary for the computer calculations.

Under the gradient zone we have a situation where the disturbances from the outlet are less than those from the inlet above the gradient zone. The velocity level is lower, but we still have turbulent fluctuations. The velocity field must be determined from experiments.

Near the wall the natural convection phenomena are dominating.

Natural convection along a vertical isothermal plate and vertical plates with constant heat flux are treated in the literature.

In a water heat storage the situation is more complex.

- During charging, neither the wall temperature nor the heat flux is constant.
- The characteristic length in the Gr-number is not constant.
- The boundary layer character can be both laminar and turbulent.
- Near the inlet there is a great influence of forced convection.
- There is a strong temperature stratification in the water.

Non dimensional analyses gives qualitative information about

- The boundary layer character (3.23).
- Forced or natural convection (3.25, 3.26).
- Dependence of Gr-number (3.12, 3.16).
- Heat transfer relations (3.8, 3.20).

The boundary layer theory gives information about

- Laminar velocity profile (3.21).
- Laminar boundary layer thickness (3.22).
- Turbulent velocity profile (3.24).
4. STORAGE EFFICIENCY

The overall efficiency of a system containing a heat storage may be a complicated function of all the component efficiencies. The efficiency of the storage is affected by the other components in the system and should be determined once the application is known.

Looking only at the content of heat the only losses are those to the surroundings. Still, as shown in chap. 1, the energy could be at such a temperature that it is not directly useful in the system, but requires an extra supply of heat, if this is available, before using. This is clearly a drawback but cannot be correctly evaluated until the cost of this extra heat is known.

It is also important to maintain a high temperature level to enable a high heat power output from the storage.

Nevertheless there is a need of defining an efficiency, based on energy, that could be used as a mean of comparing different storages, without knowing the application.

Lin, Sha and Michaels [20] suggest a way to define one efficiency for charging and one for discharging, based on energies only. These would then give the total efficiency as their product. That is

discharge efficiency

\[ \eta_{en,d} = \frac{\text{actual heat output}}{\text{ideal heat output}} \]  

(4.1)

charge efficiency

\[ \eta_{en,c} = \frac{\text{actual heat reserve}}{\text{ideal heat input}} \]  

(4.2)

and the total efficiency

\[ \eta_{en} = \eta_{en,c} \cdot \eta_{en,d} \]  

(4.3)

The real value of maintaining a high temperature level, that is keeping the quality of the heat, is not known until the application of the storage is known. Still there is also here a need of defining a neutral way of comparing...
storages in this sense. One way is to calculate the useful part of the energy as defined by the exergy.

The **exergy** is defined as that part of the energy that is fully convertible into all other forms of energy. [2]

For heat this would be the part of the energy which is convertible into work. If the heat is available at a constant temperature, this part could be determined by the Carnot efficiency according to

$$ E = \eta_{Ca} \cdot Q = \left(1 - \frac{T_{ref}}{T}\right) \cdot Q $$ \hspace{1cm} (4.4)

where

- $Q$ = heat
- $E$ = exergy of the heat
- $T$ = actual temperature for the heat
- $T_{ref}$ = an available reference temperature of a source that could be used as a heat sink in a Carnot-cycle.

If the heat can only be used by decreasing the temperature of the medium down to the reference temperature, the content of exergy would be calculated as [2]

$$ E = m \cdot c \cdot \left[T - T_{ref} - T_{ref} \cdot \ln\left(\frac{T}{T_{ref}}\right)\right] $$ \hspace{1cm} (4.5)

where

- $m$ = mass
- $c$ = specific heat

(4.5) is the formula applicable for a sensible heat storage.

The correct choice of the reference temperature is clearly dependent on the actual application, the lower this value is the higher will the exergy part of the energy be.

For space heating a normally used value for $T_{ref}$ is $20^\circ$C. A choice of $T_{ref}$ equal to the return temperature from the network to the storage is also reasonable to use, since temperatures below this value are not useful.
Of course the exergy should not be seen as the actual content of useful energy for heating, but merely be used as a mean to compare different storages and different experiments and thereby account for the temperature level of the energy.

The definition of exergy efficiencies would be similar to those for energy, namely for

\[ \eta_{ex,d} = \frac{\text{actual exergy output}}{\text{ideal exergy output}} \quad (4.6) \]

and charging

\[ \eta_{ex,c} = \frac{\text{actual exergy reserve}}{\text{ideal exergy input}} \quad (4.7) \]

There are many other definitions for the efficiencies but those above are used in this paper.
5. BACKGROUND

A lot of work has been done on the subject of heat storage during the last 10-20 years, especially on heat storages in solar applications. Experimental studies on laboratory plants as well as on large scale ones have been done. Theoretical studies on the internal behavior of a storage as well as of systems containing storages have been performed.

We will here discuss some of these works without the intention of giving a complete survey. Most of the investigations are based on temperature measurements and just a few of the published papers covers velocity measurements.

Ada, Striebel (1981) [1] presents a one-dimensional model that includes a factor $\varepsilon$ to calculate the influence on the thermocline from heat losses at the top and the wall above the gradient zone. This factor take care of the convection induced by these heat losses. Their model is validated by experiments in a 4.5 m³ storage tank and a good agreement is achieved by matching this parameter. Their model was developed to simulate a seasonal heat storage.

Fritzsche (1985) [10] made comprehensive measurements on a 2000 m³ cylindrical steel vessel integrated in a district heating system. He presents results for the heat losses, temperature stratification, influence from the inlet and the charging volume flow on the formation of the gradient zone. He has used the model of Ada, Striebel and can show good agreement between experiment and simulation, for an adequate choice of $\varepsilon$.

Straub, Grigul (1977) [28] made numerical, experimental and analytical studies of the influence from convection in a seasonal heat storage. They used a two-dimensional numerical model to calculate the development of the thermocline in a seasonal storage. One of the conclusions was that such an advanced model demanded to much CPU-time to be used for the analyze of a complete year cycle.

Sliwinski et.al (~1979) [27] defines a way to calculate the temperature gradient in the thermocline as the quotient of the temperature difference and the vertical distance between the two points where the temperature gradient gets less than 10% of its maximum value. Sliwinski relates the
initial penetration depth for the incoming water to the Richardson number (Ri). The definition of Ri is based on the inlet velocity, the distance between inlet and outlet and the average temperature difference between the inlet and the initial temperature in the storage. He gives a critical value for the Ri-number below which mixing becomes strongly increased with decreasing Ri-number. He also shows the influence on the temperature gradient from the Ri-number for different Pe-numbers.

Wu (1978) [32] studied the temperature stratification in a rectangular tank connected to a solar collector system. He studied four different operation conditions, static conditions, charging with the collector pump, discharging with the hot water pump and automatic system operation. He found that a sharp thermocline can be maintained but the degree of stratification depends strongly on the tank configuration (height to diameter ratio), inlet and outlet port design and location, mass flow, velocities and the temperature difference. Two new mathematical models were developed for incorporation into the TRNSYS-code. A modified viscous-entrainment model which represents a more realistic storage tank performance and another model that takes care of turbulent mixing and a heat exchanger in the storage.

Guo and Wu (1981) [12] developed a two dimensional time dependent model based on the principles of natural and forced convection. They present results for arbitrary Re- and Gr-numbers for conditions in the laminar region.

Lin, Sha and Michaels (1979) [20] defines the thermal energy storage efficiency (TESE) for a stratified storage tank, including the discharge efficiency and charge efficiency. They also define the overall system efficiency. They developed the COMMIX-SA code, a three dimensional thermo hydrodynamic code to investigate the flow stratification phenomena.

Lin and Sha (1979) [21] Making use of COMMIX-SA they simulated different types of baffles inside the storage tank. They found vertical baffles more effective than horizontal, aspected height to diameter ratio was three or four. A tall cylindrical tank with vertical concentric cylindrical baffles and a ring distributor can provide discharge and charge efficiencies of 90% or higher.

Oppel, Ghajar, Moretti (1986) [24] developed a one dimensional explicit finite difference model for a stratified storage tank. The model was tested on
published experimental data. The model covers through-flow conditions for charging or discharging the tank, conduction and turbulent mixing simulated through thermal eddy conductivity factors determined from experimental data. They concluded that the mixing in the tank is dependent on the Re- and Ri-numbers and the inlet configuration. In order to obtain the dependence numerous simulations of various experimental data must be examined.

Oppel, Ghajar, Moretti (1987) [25] completed the model with a decreasing hyperbolic function to predict the eddy-conductivity factor. A general relationship between the inlet eddy-factor and the ratio of Re- over Ri-numbers was established for the inlets investigated. Nine different experiments were examined for various flow rates (Re-numbers) and Ri-numbers. All the Re/Ri ratios fitted a straight line in a log-log diagram vs the eddy factor. Good agreement between the simulation model and the experimental data was found, however further examination of experimental data is needed.

Cohen, Callaghan (~84) [7] studied different types of port configurations over a wide range of flow conditions in a cylindrical steel vessel with L/D = 1.44 and a volume of 2.44 m³. They studied vertical and horizontal inlet geometries and a vertical distributor. They used the relation suggested by Turner [29] for turbulent jets with reversing buoyancy to calculate the depth of penetration. A one dimensional finite difference model was suggested and good validation was provided to the measured thermal effectiveness.

Kandari, Moustafa and Marafie (?) [16] studied the L/D ratio for five different storages during stand still periods, and found that the extraction efficiency increased by increasing L/D ratio. A L/D ratio over four is undesirable due to the extraction efficiency.

Hess and Miller (1982) [13] studied natural convection flow in a cylindrical enclosure with laser doppler anemometry. They found that the wall could have a strong effect in destroying the thermocline. Both radial and axial components were measured for Ra-numbers between $3.7 \cdot 10^7$ and $7.5 \cdot 10^7$. The velocities measured ranged between 0.01cm/s and 0.45cm/s. The tank used had a height of 38 cm and was made of aluminum (conductivity = 150 w/mK). Velocity measurements could be done as close as 0.6 mm from the wall. Very good agreement was found with an numerical solution.
Cole and Bellinger (1982) [8] presented a one-dimensional analytical model of a stratified tank, experimental measurements of thermal stratification in five different tanks, correlation of experimental data with empirical constants in the analytical model and a procedure for designing thermally stratified tanks. They define a stratification index that allows comparison between different tanks. They found that for Ri-numbers less than 0.5 the stratification index drops off sharply.

Lavan and Thompsson (1977) [19] made experiments with different L/D ratios and different inlet geometries and found that the stratification improves with increasing L/D, AT and inlet, outlet, port diameter and it decreases with increasing flowrates. A ratio between 3 or 4 for L/D seems to be reasonable.

Yoo, Wildin and Truman (1986) [35] performed experiments on a scale model thermal storage to study the initial formation of the thermocline by means of flow visualizations. The inlet densimetric Froude number was found to be a governing parameter in the formation of the thermocline. A Fr-number greater than about 2 was found to give good stratification. Some recommendations are made on the construction of inlet diffusors.

The listed works covers most of the field of water heat storing, and there are even more that could be mentioned. Both experimental and theoretical works are presented, and are in many cases dealing with solar applications for water heat storages.

A lot of models have been developed that gives good predictions for the experiments they are tested against, but they are normally not validated by tests on many storages.

Numerical models that are either one-, two- or fully three-dimensional exists. The one-dimensional models requires some simplified theory to handle the convection in the storage, induced by the inertia of the incoming water or by the heat losses. This is normally handled by introducing parameters that has to be choosen from experience from previous experiments.

The two-and three-dimensional models ought to give good results for cylindrical storages if the scale of the calculation grids could be choosen according to the scale of the phenomenas that are involved. This would however de-
mand a very fine grid for example in the boundary layer at the wall and near the inlets. The scale in the turbulence is certainly also sometimes very small and will probably require some kind of turbulence model involved in the program.

To our knowledge these models still requires to much CPU-time on very powerful computers to be widely used for calculations on a whole cycle for a storage. They are however valuable tools to study single phenomenas involved.

Most of the heat transport phenomenas that are involved in a water heat storage have been experimentally studied to some depth. Very accurate studies are performed on single phenomenas and under controlled conditions. There are however very few works presented on velocity measurements in heat storages.

An approach, where all the involved phenomenas and their interactions are integrated, is still lacking. This state of the art could be sufficient when single phenomenas are dominating, but will certainly not cover all applications for heat storing.

Our aim is to get a better knowledge about the actual phenomenas by doing comprehensive measurements of temperatures and velocities in a storage, where they are all involved. The measurements are performed in a medium sized steel vessel and further on a small sized plexi-glass model will be used.

This could give a possibility to determine which phenomenas that are dominant under different conditions and thereby get a mean to determine the value of the parameters that are necessary in a simple model.

The final goal is to achieve a model that covers the important phenomenas and still is possible to use on small computers in engineering work.
6. EXPERIMENTAL SETUP

The work started with the construction and building of a laboratory pilot plant to supply the capacity for heating during charging and cooling during discharging. Fig. 6.1 shows a drawing of the main parts in this plant.

There are two pumps, one for charging and one for discharging, each one with a frequency regulator to enable a smooth varying of the flow. The two boilers have a heating capacity of 18 kW each and can be regulated by thyristors from 1 to 18 kW. The thyristors 45 A/500 V are of zero passing type and are opto-coupled to the control system (Billman/Landis & Gyr, Visogyr), which by a voltage 0-10 V to an electronic circuit can vary the pulswidth of the signal to the thyristors and thereby the power output.

The boilers are heated to a temperature of about 10-20°C above the pre-selected charging temperature. To avoid stratification in the boilers, it was necessary to install an extra pump to circulate the water through the boilers and keep them well mixed.

The correct charging temperature is then achieved by mixing water from the boilers with return water from the storage in a 3-way automatic control valve regulated by the control system, which gets the temperature from a temperature gauge after the valve. A water flow meter gives together with temperature gauges in the feed water and in the return water pipe the necessary signals to a heat supply meter. The hot water is supplied at the top of the storage during charging.

During the discharging the hot water is withdrawn from the top and passes a heat exchanger, which on the secondary side uses tap water to cool the water from the storage down to a preselected temperature, by varying the water-flow on the secondary side. A temperature gauge in the pipe for return water to the storage gives a signal to the control system which regulates an automatic control valve in the tap water pipe. The return water is then supplied at the bottom of the storage.
To take care of the change of volume of the water in the system, when the temperature is varying, there are two expansion vessels. These keep a constant pressure in the system since they are coupled to the pneumatic system in the laboratory, through an automatic pressure control valve.

To avoid contamination of the heat exchanger and to get clean water in the storage there is a filter in the system which demands a lot of piping, since the flow through it must be in the same direction during charging and discharging. The water in the system has also passed a desalination plant in order to get a good quality of the water and a smaller risk for contamination and corrosion in the system.

The temperatures for the control system are measured with resistance gauges of PT 100 type.

Fig. 6.2 shows the storage which is cylindrical with the diameter 0.8 m, the height 2.4 m and the total volume 1.2 m³. The storage has twelve windows to enable visualizations and measurements with Laser-Doppler-Anemometry and with Particle-Image-Velocimetry technique. The storage is insulated with 7 cm of mineral wool, giving an overall heat transfer coefficient of 0.6 W/m².

The inlets at top and bottom are of the same type and the one at the top is shown in fig. 6.3. The water comes in to the storage and is spread in horizontal and radial direction, at the top of the storage during charging and at the bottom during discharging. The annular slots at the inlets are adjustable from 1 to 33 mm to make it possible to get different inlet velocities without changing the flow rate.

The data acquisition system consist of a micro computer Victor Sirius-1 for storing and simple processing of data. The computer has a HPIB-interface for communication with the MIKROLINK converting system provided with a 12-bit analogue to digital converter, units for handling signals from thermocouples and resistance gauges, counting units and digital output for controlling a scanner. To this Solartron Analogue Scanner are then all the thermocouples connected, a total amount of 56 gauges.
Figure 6.2  The storage with the levels for measurements of the horizontal temperature distribution.

Figure 6.3  The inlet at the top of the storage.
The final processing of data is performed on a MIRA AT personal computer using software PC-MATLAB [23].

Temperature measurements done with the data acquisition system are all performed with thermocouples of the Copper-Constantan type. All the thermocouples has a common cold junction, in an insulated box, and the temperature of this is measured with a resistance gauge.

During the experiments data from 49 temperature gauges in the storage are stored in the data system. The temperature gauges are located at 24 different levels and at four levels there are six to eight gauges in the same horizontal plane, and at different distances from the wall, to measure the horizontal variation of temperatures. A drawing is shown in fig. 6.2.

All the temperature measurements done with the data acquisition system are performed with gauges that are individually calibrated. For the calibration we used a temperature bath (Heto) with a very accurate control of the temperature, variation less than 0.05°C. The temperature of the bath was measured with a temperature gauge of resistance type (Chinon PT 25) and using an accurate DVM (Solartron 7075) giving an error of the measured temperature of less than 0.01°C.

The gauges, that were connected to the acquisition system, were then placed in the bath and their voltage output, as registered by the acquisition system at different temperatures in the actual range, were measured and stored. From these values we arranged the gauges in four groups and used four different polynomials for converting voltage to temperature with the computer.

Every measurement of the temperature is based on the collection of 10 or 20 values of the voltage output from the gauge and the conversion to temperature is based on the mean values of these. By doing so we got an error less than 0.2°C, when the temperature outputs from the system were controlled against the calibration system described above. In this way we got a sampling rate of about 50 values per minute. We consider the determined temperatures to be correct within 0.5°C absolute values and within 0.3°C relative values.
Flow and heat supply measurements are performed with uncalibrated commercial flow meters (Valmet) of wing wheel type with an error less than 2% according to the manufacturer. The flow meters give pulses for every 0.25 litre to the heat supply meter and the data acquisition system. The heat supply meter gets the temperatures from PT 100 gauges, calculates the heat supplied and delivers pulses to the data acquisition system. The total error is less than 3% according to the manufacturer. Comparisons between the heat supplied to the storage, according to the heat supply meter, and the calculated heat content in the storage indicates that these figures are reasonable.

To measure the velocities a tracker based two component Laser Doppler Anemometer was used (DANTEC). For a description of the use of the LDA-equipment see e.g. [11]. In fig. 6.4 the LDA arrangement is shown. The laser is of He-Ne type with a power of 15 mW. Because of the low power and the need of having a good signal to noise ratio the LDA had to be run in a forward scatter mode. This required the photomultiplier to be mounted on the other side of the storage, which also had the consequence that the photomultiplier had to be individually traversed.

![Experimental Set Up Laser Doppler Anemometry](image)
We have used two focusing lenses, one with a focal length of 310 mm for measurements in the center of the storage and the other with the focal length 80 mm for measurements in the boundary layer at the wall. Together with the 80 mm lens we used a beam expander to get a smaller probe volume. The size of this can be estimated to about 0.3 mm \cdot 4 mm with 310 mm lens and 0.07 mm \cdot 0.15 mm with 80 mm lens together with the beam expander. However the measuring volume size, which depends on the position of the photomultiplier and the gain in the system, is probably smaller than the probe volume, as indicated by the capability of resolving velocities near the wall.

Two Bragg cells, mounted in the optical unit shift one laser beam 40 MHz and one 70 MHz. Together with an unshifted beam this gives two moving interference fringe patterns in the measuring volume, one moving horizontally and the other vertically. The photomultiplier current will have two frequencies that, depending on the direction of the flow, are either above or below the shifted frequencies.

An Apple II computer and a Nicolet frequency analyzer were used for acquisition and analyze of data, the first one for the horizontal velocities and the second for vertical velocities.

The software to the APPLE computer can handle measurements of one component of the velocities and, for a preselected amount of collected datas, show a plot of the frequencies of the original signals as well as calculate the mean value and standard deviation for the velocities from signals captured by the tracker. The Nicolet shows the spectrum of frequencies for a preselected amount of sampled datas. All the needed datas are then manually registrered and with a word processor written to a datafile for later processing with MATLAB.

The carrying through of an experiment always began by starting the appropriate pump and the boilers in the supply unit. For a charging experiment the heat exchanger had to simulate the return temperature from the storage. The heating up period continued until a stable value of the preselected temperature after the mixing 3-way valve was achieved. Then the valves to the storage were opened, the charging of the storage started and the circulation through the heat exchanger was stopped.
For a discharging experiment the temperature after the 3-way valve had to simulate the temperature at the top of the storage. The circulation in the supply unit continued until the temperature after the heat exchanger had stabilized on the preselected value for the return temperature to the storage.

After these preparations the experiment and the measurements started.
7. EXPERIMENTAL RESULTS AND DISCUSSION

Physical quantities that are measured are temperatures and velocities. Velocities in two perpendicular directions either in one fix point in the storage or in separate points while traversing in vertical or horizontal direction.

Temperatures and velocities are measured once a minute.

The magnitude of these quantities are determined by the physical limits of the experimental setup. These limits are:

\[
\begin{align*}
T_{\text{max}} &= \text{maximum temperature in the storage} \quad 90^\circ\text{C} \\
T_{\text{min}} &= \text{minimum temperature in the storage} \quad 10^\circ\text{C} \\
\Delta T &= \text{temperature difference} \quad 0.5-40^\circ\text{C} \\
q &= \text{waterflow rate} \quad 0.15-1.4 \text{ m}^3/\text{s} \\
v_{\text{inl}} &= \text{inlet velocity} \quad 0.001-1.1 \text{ m/s} \\
v_{\text{st}} &= \text{mean velocity in the storage} \quad 0.3-2.4 \text{ m/h}
\end{align*}
\]

The relations between those are

\[
\Delta T = T_{\text{max}} - T_{\text{min}} \quad (7.1)
\]

\[
v_{\text{inl}} = \frac{q}{A_{\text{inl}}} \quad (7.2)
\]

\[
v_{\text{st}} = \frac{q}{A_{\text{st}}} \quad (7.3)
\]

where \( A_{\text{st}} = \text{area of a horizontal cross section} \)
\( A_{\text{inl}} = \text{area of the inlet slot} \)

The maximum temperature is determined by the fact that the vessel is not pressurized and that it has glass windows. The temperature difference between hot and cold water cannot be maintained smaller than 0.5°C due to problems with the stability of the temperature control system. The higher limit is determined by the maximum temperature difference that is allowed along the windows.
Limits for the waterflow are due to problems with the stability of temperatures at small flows and the maximum flow is determined by the capacity of the pumps.

These are also the reasons for the limits for inlet and mean velocity in the storage.

7.1 Qualitative conclusions from temperature measurements

A typical graph showing the temperature variation for some selected temperature gauges in the storage is shown in fig. 7.1. The graph shows the variation of temperatures during charging.

![Temperature variation graph](image)

*Figure 7.1* Temperature variation for eight gauges at different levels from top to bottom in the storage. $\Delta T = 28.7^\circ\text{C}$, $q = 0.6 \text{ m}^3/\text{h}$, $v_{\text{inl}} = 0.0275 \text{ m/s}$.  

From such a graph one can draw some qualitative conclusions about the behavior of the storage.
First we should define what we call the **gradient zone** in the storage. By this we mean the volume in the storage where the temperature gradient, \( \frac{dT}{dh} \), is greater than zero.

The temperature graph in fig. 7.1 shows the temperature as a function of time, \( T = T(t) \) and the slope of this graph does not directly give \( \frac{dT}{dh} \) but instead \( \frac{dT}{dt} \). To get the **temperature gradient** from this value it must be divided by the mean velocity in the storage \( \frac{dh}{dt} \). That is

\[
\frac{dT}{dh} = \frac{dT}{dt} \frac{dh}{dt}
\]  

(7.4)

If one want to compare the results of two different experiments by comparing the slope of the temperature graph in such diagrams, one must have this in mind.

The graphs above show the result of a rather stable charging of the storage. Compare the lower part of the gradient zone, that is where the temperature starts raising in the graphs above, with the upper part where the temperatures are reaching their maximum values. One can see a great difference in the rate of change of the slope in the two parts of the gradient zone.

In the lower part of the zone the gradient reaches its maximum value much faster than it goes to zero in the upper part. This is due to the mixing of hot and cold water near the inlet at the top. When the hot water enters a storage, completely filled with cold water, the warm water will be able to penetrate the cold water down to a certain depth.

This initial **depth** is mainly determined by the temperature difference between hot and cold water and the inlet velocity of the water. The incoming water is then mixed with cold water in the storage down to this depth. The bigger this mixed volume is the slower is the growth of its temperature.

This penetration depth will then increase, as the temperature difference between the water in this volume and the incoming water is decreasing, thereby causing a slower growth of the temperature in the upper part of the gradient zone.

It is possible to draw some conclusions about the **change of the temperature gradient** during one charging sequence from a graph as the one above.
The temperature gradient is decreasing when the gradient zone is moving downwards. Since we have kept the charging rate constant during one cycle it is possible to compare the gradients at different levels in the storage.

In order to compare the development of the gradient for different charging rates, one must have in mind that the gradient in the $T(t)$ graph has to be scaled with the mean velocity to give the temperature gradient. (Eq. 7.4)

![Graph](image-url)

**Figure 7.2** Normalized temperature vs normalized time at three levels in the storage. Dots for experimental values and line for theoretical ones. $\Delta T = 16.4^\circ C$, $q = 0.382 \, m^3/h$, $v_{in} = 0.0298 \, m/s$. $t''$ is the time for the charging of one storage volume.

The plot in fig. 7.2 shows the temperature variation for three gauges in the storage. Crosses show the experimental results from one charging period and the lines shows the corresponding theoretically calculated temperatures at the same levels. The latter are calculated as if the water was moving as a plug through the storage and the temperatures were effected only by heat diffusion in vertical direction.
The time is normalized by dividing with the time for charging of one storage volume and temperatures by dividing the actual temperature difference to the minimum temperature with the total temperature difference.

Here one can see that there is a big difference between theoretical and experimental values. The difference for the first gauge, that is the one near the top of the storage, shows the effects of the mixing near the inlet as discussed above.

One can also see that the decrease of the gradient is stronger for the experimental results than for the theoretical. This is due to natural convection in the storage caused by the heat flux from the water to the wall.

This heat flux depends mostly on the heat losses through the wall, but also on the heat that is required to increase the temperature of the wall and thus will be stored in it.

Since the walls in our storage are made of steel, with a thickness of only 4 mm, their temperature will rise rather fast when they get in contact with hot water. This means that the heat to the wall is mainly taken from the gradient zone and the volume just above it during charging.

When the gradient zone is moving upwards, that is when we are discharging the storage, the walls will return the heat to the water mainly in the gradient zone and the volume just below it.

In both cases this phenomena will increase the thickness of the gradient zone. The net effect of this exchange is that the temperature level of the heat, its exergy, is decreased.

The heat losses to the surroundings causes natural convection in the whole storage. The effects of these are best seen in a graph showing the temperature distribution in a stratified storage during a stand still period, as in fig. 7.3 and fig. 7.4.

Big temperature differences to the surroundings and poor insulation gives strong convection.
Figure 7.3  Temperature vs height at different times during a stand still period after 0, 5, 15 and 21 h. Insulated storage.

Figure 7.4  Temperature vs height at different times during a stand still period after 0, 5, 15 and 21 h. Uninsulated storage.
The effects from this convection on the temperatures in the warmer part of the storage appears as a decrease of the mean temperature in this volume and also as a broadening of the gradient zone. How these heat losses are shared between the gradient zone and the volume above it is not yet quite known.

Below the gradient zone this convection causes a secondary stratification but the effect on the gradient zone is rather small. See fig. 7.3 and 7.4.

The fact that no such stratification is seen above the gradient zone is due to Benard convection caused by the heat losses at the top of the storage. This convection will mix the volume above the gradient zone.

The heat losses are of course bigger in the uninsulated case although the temperature level in the warmer part is lower than for the insulated one. There is no stratification above the gradient zone in either case.

This shows that the Benard convection is very strong as it can keep the volume above the gradient zone mixed, although the heat flux to the wall is big and should give stratification in the uninsulated storage. Stratification is achieved during the charging period, as can be seen in fig. 7.4 which shows a weak stratification above the gradient zone when the stand still period starts.

For the uninsulated storage the Benard convection during stand still seems to be more effective for the mixing, than the inertia of the incoming water during the charging period.

For the insulated storage the heat losses are so small that the inertia of the incoming water during charging prevents the volume above the gradient zone from stratifying.

Both figures show, that the temperatures in the upper part of the gradient zone are more effected than those in the lower part by convection induced at the wall. Without this convection the temperature change would be symmetric.

It is also interesting to compare the thickness of the gradient zone in these two cases. It seems to be so that the thickness is smaller after 21 h for the uninsulated storage. This is due to the Benard convection that mixes the water
in the warm volume and prevents the gradient zone from growing upwards.

Although the uninsulated storage gives much bigger heat losses and though a less heat content in the storage this does not give a faster growing thickness of the gradient zone. This shows that the growing rate for the thickness of the gradient zone is a poor measure of the quality of a heat storage.

The volume below the gradient zone is stronger stratified in the insulated storage than in the uninsulated. This is due to the small temperature difference to the surroundings in the uninsulated case. The fact that stratification appears below the gradient zone shows, that the convection cells probably have a small extension in vertical direction.

What can we say about the natural convection during a charging period? There will be no Benard-convection as the losses in this case are taking place in forced convection when the incoming water passes the roof of the storage.

![Temperature vs time for eight gauges at different levels. Show a strong stratification above the gradient zone. q = 0.15 m³/h, v_{air} = 0.0035 m/s. Uninsulated storage.](image)

Figure 7.5 Temperature vs time for eight gauges at different levels. Show a strong stratification above the gradient zone. q = 0.15 m³/h, v_{air} = 0.0035 m/s. Uninsulated storage.
Natural convection above the gradient zone will still take place and give roughly the same effects as mentioned above. Still, if the heat losses through the walls are big, as in the experiments done with the uninsulated storage, this convection will cause a secondary stratification as one can see in fig. 7.5.

This strong effect prevents the mixing caused by the inertia of the incoming water and is dominant for high Ra-numbers. [17]

Below the gradient zone this convection is still the same but the effects of it can't be seen as clear as in the stand still case since the cold water at the bottom is withdrawn from the storage.

During discharging the water comes in at the bottom and will cause mixing below the gradient zone, since the heat losses to the surroundings are too small to cause stratification here.

Above the gradient zone the natural convection will cause stratification and a broadening of the gradient zone. The heat losses at the top will take place under forced convection so there will be no Benard convection. The gradient zone will effectively stop the incoming water at the bottom from effecting this stratification.

The previous figures were the results of normal chargings of the storage. If one wants to study the influence from the different phenomenas listed in chapter 2 it is necessary to make experiments during forced charging.

By forced charging we mean a situation when we achieve a strong influence from the inlet on the thermocline in the storage. This is possible either by keeping the temperature difference between hot and cold water small or by maintaining a high inlet velocity. The inlet velocity depends on the charging rate and the area of the inlet slot.

Figure 7.6 and fig. 7.7 show the results of such a charging of the storage. This should be compared with fig. 7.1 for a normal charging.
Figure 7.6  Temperature vs time for eight gauges at different levels. Shows a charging with great mixing. \( \Delta T = 10.1^\circ C, q = 1.36 \text{ m}^3/\text{h}, v_{in} = 1.06 \text{ m/s.} \) Charging starts after 5 min.

Figure 7.7  Temperature vs height for the same experiment as in figure 7.6.
The first thing to notice is that for the forced charging the temperature of many gauges starts rising at the same time. The next thing is that the gradient \( \frac{dT}{dt} \) in this case is much smaller although it should be much bigger to give the same temperature gradient \( \frac{dT}{dh} \), because of the higher mean velocity in the storage.

The change of the slope of the temperature curves is of course also much slower, indicating a slower growth of the temperatures.

All this is the results of a bigger initial penetration depth in the beginning of the charging period. As said before this will give a large mixed volume from the start and thereby its temperature will grow slowly, when the hot water is fed into it.

The T(h)-graph shows the same thing namely, that temperatures of many gauges starts rising at the same time and are increasing rather slowly. The top of the storage does not reach its full temperature before the temperatures almost at the bottom starts rising.

Figure 7.8  Temperature vs time for six gauges at the same level.
Another thing we have not yet been looking at is the temperature variation in horizontal direction in the storage.

Figure 7.8 and 7.9 show the temperature graphs $T(t)$ for six temperature gauges at the same level but at different distances from the wall. These shows that the temperatures are the same within the accuracy of the gauges. This fact has also been stated by many other experimenters in this field.

It is only when the gradient zone is passing the gauges one can detect a difference in temperatures. This can depend on a real difference that exists for a short time, but also on a small difference in the altitude of the gauges. See fig. 7.9 for an example of this.

A temperature difference in horizontal direction represents an unstable physical condition that can be maintained only for a short time, since it will cause natural convection that equilizes the temperatures.
The temperature graphs show that the horizontal equilization of temperatures is so fast that it must be convection and not diffusion that gives the dominant contribution to it.

7.2 Quantitative treatment of measured temperatures

The conclusions above are only qualitative and rises the question of which quantitative measures should be used to compare different experiments.

We have used the following quantities

- temperature gradient
- thickness of the gradient zone
- initially mixed volume
- initial penetration depth
- Richardson-number
- energy and exergy efficiency

to compare different experiments.

7.2.1 Temperature gradient

As stated in chapter 1 a good thermal heat storage shall be thermally stratified. One measure of this stratification could be the magnitude of the temperature gradient in the gradient zone. A good storage should then have a high value of the temperature gradient, which indicates weak mixing of hot and cold water.

The natural way to compute the gradient would be to take the temperature difference between two adjacent gauges and devide it by their distance in vertical direction. Unfortunately the distance between the gauges is too big to give a good value for the gradient.

A better way to do it is to use the measured temperature variation as the water passes the gauges. We just take the temperature difference between two consecutive values for the gauge and devide it with the distance that the water has moved during the same time interval. This distance equals the mean velocity in the storage multiplied by the time interval. Eq. 7.4 gives
This way to do it is of course only possible if the liquid is moving up or down in the storage. The mean velocity must also be roughly constant.

Another restriction is that there must not be any great changes in the gradient during the time interval, which is normally one minute. This means that it is not possible to determine a value for the gradient in the mixed zone near the inlet, where the temperatures changes all the time.

Small temperature differences between hot and cold water means that the temperature difference between two consecutive measurements is comparable to the inaccuracy for the temperature gauges. This will make it difficult to determine the gradient in those cases.

All these restrictions is fortunately no problem if we want to compare storages under normal conditions, that is under normal charging or discharging.

An example of the variation of the calculated value for the temperature gradient is shown in fig. 7.10. Here the gradient is calculated at eight fix levels in the storage and as a function of time. The gradient is calculated when the gradient zone passes the gauge at the specified level.
Figure 7.10 Temperature gradient vs time as calculated at eight different levels for a normal charging. $\Delta T = 28.7^\circ C$, $q = 0.6 \text{ m}^3/\text{h}$, $v_{inl} = 0.0275 \text{ m/s}$.

Figure 7.11 Temperature gradient vs time as calculated at eight different levels for a forced charging. $\Delta T = 2.5^\circ C$, $q = 1.00 \text{ m}^3/\text{h}$, $v_{inl} = 0.0236 \text{ m/s}$. 
Figure 7.11 shows quite clear that the magnitude of the temperature gradient is much smaller in the case with forced charging.

Table 7.1 shows the value of this gradient for some experiments with different inlet conditions.

The first five of the values in the table above show the influence from the temperature difference between hot and cold water and the next four indicate the influence from the velocities on the temperature gradient. The tendency is quite clear, to get high values for the temperature gradient the temperature difference should be high and the inlet velocity low.

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<th>$v_{\text{inl}}$ (m/s)</th>
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Table 7.1 Calculated temperature gradient at the same level (just above the middle of the storage) in different experiments.
7.2.2 Thickness of the gradient zone

Another way to measure the degree of stratification is to determine the thickness of the gradient zone. This will also give a measure of the degree of mixing of hot and cold water in the storage.

Ideally the gradient zone is the volume in the storage where the temperature gradient is greater than zero and its thickness should be the vertical extension of this zone. This is not a practical measure to use.

To avoid influence from the inaccuracy of the temperature gauges and the statistical spread of values, one has to set a certain limit wherein this gradient zone should be considered to exist.

We have used two ways to determine the thickness of the zone. Either by using the temperatures directly or by the use of the calculated temperature gradient.

A factor $\varepsilon$ is used to set up these limits in the calculation of the thickness. When we use the gradient to determine the thickness, this $\varepsilon$ gives what fraction of the maximum gradient for that certain gauge, that should be used.

When temperatures are used $\varepsilon$ gives what fraction of the maximum temperature difference to use in the limits. That is, the thickness is the vertical extension of the volume where

$$\frac{dT}{dh} > \varepsilon \cdot \left(\frac{dT}{dh}\right)_{\text{max}} \quad (7.6)$$

or

$$T_{\text{min}} + \varepsilon \cdot (T_{\text{max}} - T_{\text{min}}) < T < T_{\text{max}} - \varepsilon \cdot (T_{\text{max}} - T_{\text{min}}) \quad (7.7)$$

The latter method is better when the temperature gradients are small.

These values for the thickness can then be calculated at different levels in the storage and for different values for $\varepsilon$, (0.05 to 0.20). An example of this is shown in fig. 7.12.
Figure 7.12  Thickness of the gradient zone as calculated at eight different levels in the storage. Four different experiments and two theoretical cases.

The value for this thickness, as well as the development of it as the gradient zone moves down through the storage, could then be compared to an ideal value. This value is calculated as if the water was only affected by heat diffusion in vertical direction during its way down through the storage.

The thickness is here calculated at 8 different levels as the water moves down the storage, with the first method and for $e = 0.15$. All curves are results from what we call normal charging, that is during rather stable conditions.

One can see that the thickness of the gradient zone as it reaches the first of our gauges is roughly the same in three of the experiments. The further development of the thickness for those shows a bit faster growing than for the theoretically calculated. This indicates the influence from the convection that takes place in the storage.

One of the curves differs from the others. The charging rate is here the same as in the other experiments but the stabilizing temperature difference is
smaller, causing a larger thickness of the gradient zone in the beginning of the charging period. As the zone moves downwards, the thickness does not even grow as fast as in the theoretical case.

This could depend on an error in the measured values, but is probably due to mixing induced from the inlet. This mixing can give a decrease in the thickness of the gradient zone although it represents a poorer stratification.

The remarks above show that the thickness of the gradient zone is not always a correct measure for the quality of the stratification. Table 7.2 shows the calculated values for this thickness in some experiments. The thickness is given at a fix level in the storage, 1.9 m from the top, $s_h$ and after a fix time, 50 minutes of charging, $s_t$.

It is difficult to calculate exact values for the thickness of the gradient zone, but the table still indicates the influence from temperature difference and inlet velocity on this thickness. High temperature differences and low inlet velocities gives small values for the thickness of the gradient zone.

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Table 7.2 Thickness of the gradient zone, at a fix level and at a fix time during a charging period, in different experiments.

7.2.3 Initially mixed volume, initial penetration depth
The first part of a charging or discharging period is very important in determining the thermocline in the storage.

During this initial part the starting value for the temperature gradient and the thickness of the gradient zone is determined. After this establishing of the zone it will mostly be effected by heat diffusion and natural convection and only to a small extent by forced convection, induced from the inlet.

Since this initial mixing in the storage is so important, it seems natural to define an initially mixed volume or an initial penetration depth, as a mean to compare different experiments.

The definition of these two quantities could be

\[
V_{\text{mix}} = \text{the volume that the first incoming water will mix when it enters the storage.}
\]

\[
h_{\text{mix}} = \text{the depth in the storage that the first incoming water will reach.}
\]

The relation between the two quantities will then be

\[
V_{\text{mix}} = h_{\text{mix}} \cdot A_{\text{st}} \quad (7.8)
\]

These quantities shall then be calculated from the measured values of the temperatures.

The mixing in the storage gives as one result that the heat front have moved further down in the storage than is predicted by theory. \(V_{\text{mix}}\) can then be calculated from the temperature measurements for the different gauges.

From these measurements one get the time when the heat front reaches that gauge in the experiment. The charging rate, together with this time, will then give the volume that is fed into the storage.

Since the volume above each gauge is known this makes it possible to calculate the initial mixing volume and the penetration depth as

\[
V_{\text{mix}} = \text{volume above the gauge} - \text{actually charged volume} \quad (7.9)
\]
This is based on the assumption that the front shaped at the initial penetration depth will continue down the storage with the mean velocity of the water.

The inaccuracy of the charging temperature will give variations in the penetration depth and make it a bit difficult to determine an accurate value for this depth.

One way to get better values for this depth could be to calculate a value for it based on many temperature gauges. Unfortunately the time lag between these measurements means that natural convection and heat diffusion have affected the heat front.

The best way to do it seems to be to take a mean value for this penetration depth, calculated from the registered temperature variation for a few gauges, just below the initially mixed volume. Our calculation is based on the time variation of temperatures for two or three gauges with 0.3 m vertical separation.

The detection of the time when the heat front reaches a gauge can be based on the temperature variation directly or on the variation of the temperature gradient.
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<td>0.045</td>
<td>118</td>
</tr>
<tr>
<td>27.5</td>
<td>2.5</td>
<td>0.024</td>
<td>43</td>
</tr>
<tr>
<td>26.4</td>
<td>2.6</td>
<td>0.014</td>
<td>65</td>
</tr>
<tr>
<td>44.8</td>
<td>1.6</td>
<td>0.043</td>
<td>134</td>
</tr>
<tr>
<td>69.9</td>
<td>4.1</td>
<td>0.032</td>
<td>47</td>
</tr>
<tr>
<td>52.3</td>
<td>16.6</td>
<td>0.311</td>
<td>78</td>
</tr>
<tr>
<td>51.8</td>
<td>16.4</td>
<td>0.030</td>
<td>92</td>
</tr>
<tr>
<td>50.2</td>
<td>28.7</td>
<td>0.028</td>
<td>42</td>
</tr>
<tr>
<td>69.0</td>
<td>24.0</td>
<td>0.024</td>
<td>36</td>
</tr>
<tr>
<td>68.7</td>
<td>24.3</td>
<td>0.014</td>
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</tr>
<tr>
<td>70.7</td>
<td>25.2</td>
<td>0.014</td>
<td>31</td>
</tr>
<tr>
<td>67.5</td>
<td>28.1</td>
<td>0.004</td>
<td>42</td>
</tr>
</tbody>
</table>

Table 7.3 Initially mixed volume for different inlet conditions.

The influence on the mixing volume, from temperature difference and inlet velocity, is indicated in the first five respectively next six lines.

### 7.2.4 Richardson number

It is difficult to draw any conclusion from table 7.3 above about the influence of charging rate, temperature difference, inlet velocity and so on, on the initial penetration depth.

Inertia of the incoming water is responsible for the mixing near the inlet and buoyancy forces are stabilizing. It is then natural to use the ratio between those two forces, the Richardson number, as an adequate quantity to relate to the penetration depth.

Normally the Ri-number is used in stratified shear flow when the density and velocity gradients are established and known.
In our case we want to use this number in a situation where these are neither established nor known.

Therefore we use a modified Ri-number defined as

\[
R_i = \frac{g \cdot \beta \cdot \Delta T \cdot h}{v^2}
\]  

(7.11)

where

- \(g\) = acceleration of gravity
- \(\beta\) = coefficient of volumetric expansivity
- \(\Delta T\) = temperature difference between incoming water and the water in the storage
- \(h\) = a typical length for the problem
- \(v\) = a typical velocity for the problem

This is a definition that for example is used by Cole [8] in his analyze of experiments on heat storage. He uses the inlet velocity as the typical velocity and the distance between inlet and outlet as the typical height.

Of course it is the inertia of the incoming water that is responsible for the mixing. Still it is not quite obvious that the inertia should be determined at the annular slot, where the water enters the storage, that is to use the inlet velocity.

The reason for this is that the velocity of the incoming water will decrease as it is spread in the top of the storage and before it starts penetrating downwards. The true value to use for the velocity must then also depend on the diameter of the storage.

One way to account for this could be to use \(Re/Ri\) as the relevant quantity to relate to the penetration depth. Reynolds number should then be determined for the mean values in the storage and would then account for the diameter of the storage. This quantity is for example used by F.J. Oppel et.al. [25]. So far we have used inlet velocity for \(u\) in the Ri-number.

What is then the typical length, \(h\), for the problem? As said before Cole and others have used the vertical distance between inlet and outlet in the storage.
This has evidently the drawback that it relates the Ri-number to a certain storage. Storages that have the same inlet conditions and the same diameter but different heights, would then give the same penetration depth for different Ri-numbers. This implies that this distance is not the correct length to use. In order to get \( h \) independent of a certain storage we have used the unit length one meter as the value for \( h \) in our Ri-number.

Another way could be to give \( h \) the value of the height corresponding to a unit volume in the storage. This way to do it would then also account for the diameter of the storage and perhaps withdraw the necessity to correct the value for \( u \) according to the diameter.

\[
\begin{array}{ccc}
\text{Ri} & \text{V}_{\text{mix}} \text{ (lit)} & \text{Re/Ri} \\
0.001 & 726 & 507440 \\
0.004 & 445 & 226150 \\
0.006 & 399 & 118780 \\
0.008 & 392 & 130580 \\
0.012 & 355 & 58601 \\
0.013 & 434 & 59972 \\
0.027 & 314 & 29084 \\
0.028 & 284 & 11377 \\
0.695 & 78 & 423 \\
3.593 & 134 & 369 \\
4.698 & 118 & 290 \\
11.441 & 43 & 45.2 \\
23.250 & 47 & 60.9 \\
31.672 & 65 & 9.53 \\
74.589 & 92 & 3.74 \\
128.450 & 42 & 2.96 \\
215.410 & 36 & 4.19 \\
602.130 & 24 & 0.89 \\
637.510 & 31 & 0.86 \\
10654.000 & 42 & 0.012 \\
\end{array}
\]

Table 7.4 Initially mixed volume related to Ri and Re/Ri.

To make a full investigation of how to choose the correct values for \( h \) and \( u \), one has to do experiments on storages with different diameters, which we have not done so far. An investigation of this type has been done by Lavan and Thomson [19].
If one relate the penetration depth or the mixing volume to this Ri-number, one get table 7.4 and the graphs in fig. 7.13 or fig. 7.14. Fig. 7.15 show how the penetration depth is related to Re/Ri.

Fig. 7.14 and table 7.4 indicates that the mixing volume is only weakly dependent on the Ri-number for Ri ≥ 0.2. For lower Ri-numbers the volume is proportional to $1/\sqrt{Ri}$. To see the value of using Re/Ri instead of Ri only, one has to do experiments on other storages too.

Figure 7.13 Initially mixed volume vs Ri-number.
Figure 7.14 Initially mixed volume vs natural logarithm of Ri-number.

Figure 7.15 Initially mixed volume vs the natural logarithm of Re/Ri.
7.2.5 Energy and exergy efficiency

Another way to compare different experiments and also different storages is to use an energy or an exergy efficiency. We shall here look at some examples on this and compare two experiments in this way. Energy and exergy is calculated in two ways. The ideal case which would be a storage that is perfectly stratified during the whole charging period and the real case where the energy content of the real storage is calculated during the whole charging period.

For the calculation of the real content the storage is divided into 25 slices each with a vertical extension between two levels for the temperature gauges. Every minute the mean temperature is calculated for every slice. The spacing between the temperature gauges is too big to give a good value for the mean temperature of the water volume between them, only by taking the mean value of their temperatures.

Instead we calculate the temperature for a certain slice by calculating the mean temperature for the water volume in the slice, when the volume passes the gauge above and the gauge below the slice. In this way we get a number of temperatures in the slice, how many is dependent on the charging rate but it will be at least three.

The real energy content, \( Q_r \), is then calculated for every slice and the results are added together to give the total energy content every minute during the charging cycle. The temperature difference in this calculation is between the real temperature of the slice and the minimum temperature in the storage, which is considered to be constant during the charging cycle. That is

\[
Q_r = \sum V_{sl} \cdot \rho_{sl} \cdot c_{sl} \cdot (T_{sl}-T_{min})
\]

The ideal energy content, \( Q_{max} \), is calculated every minute by using the heat capacity flow and the temperature difference between the incoming water and the minimum temperature in the storage.

\[
Q_{max} = q \cdot \rho_{sl} \cdot c_{sl} \cdot (T_{max}-T_{min}) \cdot t
\]

where \( t \) is the elapsed time.
In order to calculate the exergy content of the storage one has to choose the reference temperature. The right choice of this is a delicate problem as discussed in chapt. 4. We will show the result of two different choices of this temperature namely the minimum temperature in the storage and a temperature of 20°C, which is often used when the heat will be used for space heating.

If this used temperature is below the minimum temperature in the storage we correct for the exergy of the water at minimum temperature. (5.5) gives for both our cases for the real content of exergy, \( E_r \),

\[
E_r = \sum V_{si} \cdot \rho_{sl} \cdot c_{sl} \cdot \left[ T_{si} - T_{min} - T_{ref} \cdot \ln \left( \frac{T_{si}}{T_{min}} \right) \right]
\]

and for the maximum content of exergy, \( E_{max} \),

\[
E_{max} = q \cdot \rho_{sl} \cdot c_{sl} \cdot \left[ T_{max} - T_{min} - T_{ref} \cdot \ln \left( \frac{T_{max}}{T_{min}} \right) \right]
\]

The energy and exergy efficiencies are then calculated as the quotient between the real and the ideal values, when one storage volume is fed into the storage.

\[
\eta_{em} = \frac{Q_r}{Q_{max}}
\]

\[
\eta_{ex} = \frac{E_r}{E_{max}}
\]

Figure 7.16 shows the energy content during what we call a normal or stable charging period and figure 7.17 and 7.18 shows the exergy content if the reference temperature is chosen as \( T_{ref} = T_{min} = 40°C \) and \( T_{ref} = 20°C \) respectively.

The difference between the two energies in fig. 7.16 is mainly due to the heat losses but at the end of the charging period there is a decrease in the growing rate for the real energy content in the storage. This is due to the fact that warm water is now leaving the storage because of the broadening of the gradient zone. In the ideal case the charging rate is constant until the storage is completely charged and would then become zero, since the temperature of the water that leaves the storage equals that of the incoming water.
The value of the energy efficiency is here mainly determined by the heat losses and only to a small extent by the mixing at the inlet, when the charging started.

The exergy content is dependent on the choice of reference temperature and decreases with increasing reference temperature. This variation of exergy content with reference temperature has of course no meaning for the practical application of the storage. It is only a way to set a higher value on energy of a higher temperature level and shall be used only as a way to compare different experiments and different storages.

Exergy efficiency will also depend on the choice of reference temperature which can be seen in fig. 7.17 and 7.18. The lower the reference temperature is, the higher will the exergy efficiency be.
Figure 7.17  Ideal and actual content of exergy vs time with $T_{ref} = 40^\circ$C. The same experiment as in fig. 7.16. $\eta_{ex,c} = 0.922$.

Figure 7.18  Ideal and actual content of exergy vs time with $T_{ref} = 20^\circ$C. The same experiment as in fig. 7.16. $\eta_{ex,c} = 0.945$. 
Results from the same calculations performed on another experiment, with more mixing at the inlet are shown in figs. 7.19-22. The temperature difference between hot and cold water is smaller here so that the stored amount of energy will be smaller. The temperature difference to the surroundings is also smaller here, causing smaller heat losses.

One significant difference between the graphs for the actual content of energy for the two experiments can be seen at the end of the charging cycle. The growing rate for the energy content starts decreasing much earlier in the latter experiment. This is caused by the mixing at the inlet, which gives a much broader gradient zone and thereby a heat front that is far beyond the level it would have had in the ideal case with perfect stratification.

This has the effect that the water that leaves the storage at the end of the charging cycle has higher and higher temperature, thereby decreasing the growing rate for the energy content in the storage, which gives a smaller efficiency.

![Graph](image)

Figure 7.19  Ideal and actual energy content vs time. $T_{\text{max}} = 33.3^\circ C$, $T_{\text{min}} = 28.2^\circ C$, $q = 1.37 \text{ m}^3/\text{h}$, $v_{\text{st}} = 0.77 \text{ mm/s}$, $v_{\text{inl}} = 1.07 \text{ m/s}$, $Ri = 0.013$, $\eta_{\text{en,c}} = 0.862$. 
Figure 7.20  Ideal and actual content of exergy vs time with $T_{\text{ref}} = T_{\text{min}} = 28.2^\circ\text{C}$. The same experiment as in fig. 7.19. $\eta_{\text{exc}} = 0.739$.

Figure 7.21  Ideal and actual content of exergy vs time with $T_{\text{ref}} = 20^\circ\text{C}$. The same experiment as in fig. 7.19. $\eta_{\text{exc}} = 0.832$. 
The reduction of temperature level and thereby exergy with mixing becomes more evident if one choose the reference temperature more near the temperature level in the storage which a comparison of fig. 7.20 and 21 shows.

It is clear that both energy and exergy efficiencies are relevant measures of the quality of a storage. Exergy is the better one to use to account for the temperature level of the heat and the degree of stratification. For the case with poor mixing we got the values $\eta_{en,c} = 0.953$, $\eta_{ex,c} = 0.922$ and $0.945$ for reference temperature $40^\circ C$ respectively $20^\circ C$. For the heavily mixed case we got the values $\eta_{en,c} = 0.862$, $\eta_{ex,c} = 0.739$ and $\eta_{ex,c} = 0.832$ respectively.

To compare the degree of stratification caused by the mixing at the inlet during the charging period it is practical to use the quotient between the exergy and energy efficiency calculated during the whole charging period.

Figure 7.22 shows the results of such calculations for the two experiments discussed above. The energy efficiency is high at the beginning of the charging process since the difference between charged energy and energy in the storage depends on heat losses which are rather slow.

Figure 7.22  The quotient between exergy and energy efficiencies for the two experiments shown in fig. 7.16-18 and fig. 7.19-22.
The exergy efficiency is instead low at the beginning since the mixing at the inlet causes a loss of temperature level of the energy. The bigger this mixed volume is the slower is the growth of its temperature and thereby the growth of the quotient between exergy and energy efficiency.

Instead of comparing the actual content of heat with an ideal one it is possible to use the actually charged energy, as measured by the heat supply meter, which is shown in fig. 7.23. The measurements have continued for a while after the theoretical charging time. The heat content in the storage would reach a constant maximum value and the supplied heat a constant growing rate, which is needed to compensate for the heat losses from the storage, that is if the charging continues long after one theoretical charging period.

This is one way to determine the heat losses and thereby an overall heat transfer coefficient for the walls of the storage.

![Graph](image)

**Figure 7.23** Measured supplied heat and actual heat content vs time. The same experiment as in fig. 7.16-18.
7.3 Evaluation of the velocity measurements

Velocities are measured both in the boundary layer near the wall and in the center of the storage. The measurements were performed in a fix point or during traversing of the measuring point. With the present LDA-system it is possible to measure two components of the velocity, namely in vertical and tangential direction in the storage.

The natural convection at the walls is one of the reasons for the broadening of the gradient zone and therefore the velocities near the wall are interesting to study. In order to know how the heat losses are shared between the gradient zone and the rest of the storage one has to know the velocities in radial direction too.

As stated before, the mixing near the inlet is very important in determining the thermocline in the storage. Therefore it is necessary to determine the velocities in the inner part of the storage, the core. This is done either in one fix point during charging and when the gradient zone is passing the measuring point, or in different points by traversing the front lens of the LDA.

It is also necessary to know these velocities in order to determine the lengthscale for the turbulence in this volume. This lengthscale determines the rate of loss of kinetic energy that the incoming water will undergo, also in the absence of a temperature gradient.

There are some difficulties with the use of the present LDA-equipment. One is that the photomultiplier can not be mounted on the same traversing equipment as the front lens. This has the consequence that the photomultiplier has to be separately traversed and focused every time the front lens is traversed.

Another drawback is that it is difficult to measure velocities in the gradient zone due to the fact that different temperatures gives different refracting index for the water. This means that the three beams from the front lens do not have the same path length and therefore they do not give a distinct measuring volume. Furthermore the focusing of the photomultiplier is very difficult from the same reason.

The first one of these drawbacks can be handled by using a stronger laser in backscatter mode. Such a laser will soon be available.
The other one is not possible to solve with LDA-technique. Therefore we have planned to use PIV (Particle Image Velocimetry) for the measurements in volumes with varying refracting index. This technique has also two other advantages in our case.

First it gives an instantaneous picture of the velocity components in one plane and not in just one point and secondly it makes it possible to measure the third component of the velocity namely the radial one.

This technique is being continuously developed at the Division of Experimental mechanics at Luleå University of Technology.

### 7.3.1 Measurements in the boundary layer

Figure 7.24 and 7.25 shows the result of the temperature and velocity measurements during a charging period for the storage. The velocities are measured in one fix point in the boundary layer near the wall and where the velocity has its maximum value. The level is just below the center of the storage and the measurements are performed while the water is passing the measuring volume.

The temperatures are measured in the center of the storage and at the same level as the velocities.

The figures show that there is a strong relation between the temperature level and the magnitude of the velocities. Another thing to notice is that there is a weak relation between the magnitude of the velocity and the height of the zone with the homogenous temperature.

The temperature in the laboratory has varied between 20°C and 22°C so the temperature difference between the water and the surroundings varies from about 14°C to 30°C in fig. 7.24 and between 22°C and 38°C in fig. 7.26. The corresponding velocities has varied between 2.8 mm/s and 4.9 mm/s and between 4.0 mm/s and 6.0 mm/s respectively.

Figure 7.28 shows the relation between the temperature difference to the surroundings and the maximum velocity in the boundary layer. Fig. 7.29 shows that the velocity is roughly proportional to the square root of this temperature difference.
Theoretically the velocity shall be proportional to the square root of the temperature difference between the inside of the wall and the core of the water volume during steady state conditions. In our case we do not have steady state conditions and the heat transfer coefficient is not constant, so we cannot expect the temperature difference at the inside of the wall to be a constant fraction of the total temperature difference across the wall.

Furthermore the height of the volume with constant temperature have had different values in the different experiments. Therefore it is not surprising that the velocity is not perfectly proportional to the total temperature difference. Still it indicates that the maximum velocity in the boundary layer is only to a small extent dependent on the available height.

It is only just before the gradient zone reaches the measuring volume that the velocities are to some extent effected by this available height. That is when the height of the volume with a homogenous temperature above the measuring volume is approaching zero. This can be seen both in fig. 7.25 and 7.27 since the velocities start decreasing some minutes before the gradient zone reaches that level.

Figure 7.25 indicates a constant velocity below the gradient zone, until the gradient zone is just a few centimeters above the measuring volume. Fig 7.27 indicates that the velocity should be weakly dependent on the available height.

It is difficult to get a quite correct value for the Gr-number since the temperature difference at the wall is not known. This difference is only raised to the first power and therefore we can choose a value in the range 0.05 to 0.3°C, which is achieved in the measurements performed on this. This is also a reasonable range that can be achieved by a study of the heat losses through the wall.

Calculations of a local Gr-number based on a value of AT of say 0.2°C gives values that varies during the loading period, since the distance from the measuring volume to the gradient zone varies and this distance should be used as the relevant length in eq. 3.12. When the experiment starts the height above the measuring volume is 1.2 m and when the velocities in the boundary layer starts decreasing this height is 0.07 m in fig. 7.25 and 0.08 m in fig. 7.27.
Gr-numbers based on these figures gives values of $2 \cdot 10^9$ and $4 \cdot 10^9$ in the beginning of the charging period and $4 \cdot 10^5$ and $7 \cdot 10^5$ when the gradient zone gets further down and the velocities starts decreasing for the two experiments respectively.

The interpretation of this could be that the flow in the boundary layer is laminar just below the gradient zone and thereby dependent on the available height according to eq. 3.11. Further down in the boundary layer the flow might become turbulent and the velocity stops increasing with the height.
Figure 7.26 Temperature vs time at the level where the velocities are measured. \(v_{st} = 0.22 \text{ mm/s}, v_{inl} = 0.31 \text{ m/s}\).

Figure 7.27 Maximum velocity in the boundary layer vs time. See fig. 7.26.
Figure 7.24 Temperature vs time at the level where the velocities are measured. \( v_{st} = 0.22 \text{ mm/s}, \ v_{inl} = 0.30 \text{ m/s}. \)

Figure 7.25 Maximum velocity in the boundary layer vs time. See fig. 7.24.
The strongest influence on the velocities has the temperature gradient since the velocities becomes essentially zero, when the temperature gradient has its biggest value and the velocity does not reach its maximum value, until the temperature gradient is zero.

It seems to be so that no water in the boundary layer above the gradient zone passes the zone down to the volume below it. Instead the direction of the velocity for the water is changed and the water flows into the gradient zone in radial direction. Unfortunately we have not yet been able to measure the radial components of the velocities.

The velocities in the gradient zone are much higher than the mean velocity in the storage and we consider the convection to be natural.

Figure 7.28 Maximum velocity in the boundary layer vs temperature difference between the core and the surrounding air.
It is a bit difficult to resolve the velocity profile at the wall in the storage. If one use the front lens with 310 mm focal length the measuring volume will be somewhat too big to get a good resolution. The lens with 80 mm focal length gives a good resolution, but instead the measuring volume is so small that it makes it difficult to focus the photomultiplier on it.

During stand still periods it is nevertheless possible to get good measurements in the boundary layer at the wall. Fig. 7.30 shows the temperature profile in the storage during such a measurement. The measurements of the velocities were made after 16 to 20 hours of cooling down. One can see that there is a weak temperature gradient in the storage and this will to some extent decrease the velocity in the boundary layer.

Figure 7.31 shows the velocity profile at the wall and at a level 1.9 m from the bottom, that is about 0.7 m from the top.

About 0.3 and 0.4 m higher up in the storage one gets profiles as in fig. 7.32 and 7.33.
There are some differences between these profiles. Near the top the profile is broader and not as regular as a bit down in the storage. This is probably caused by the Benard convection induced at the top. Another thing to notice is that the magnitudes of the velocities are roughly the same and not proportional to the square root of the distance to the top, which is predicted in the theory for laminar boundary layers.

There could be another reason for this small change of the velocity with height in the boundary layer. During the measurements it is necessary to keep a small part of the window uninsulated. This part is about six centimeters in diameter and the velocity is measured in the middle of this area. It is not likely that this small area is totally determining the velocities so that they would be independent of the height above this point.

It is interesting to compare the experimentally determined velocity profile with a theoretical one. Fig. 7.34 shows the experimental results in fig. 7.30 but normalized. This could be compared to a theoretical laminar profile and a theoretical turbulent profile in fig. 7.35 and 7.36 respectively. The so called theoretical profiles are the usually used ones as mentioned in chapt. 3, eq. 3.21 for laminar flow and eq. 3.24 for turbulent flow.

In the laminar case should the maximum of the velocity theoretically be at a distance from the wall of about 0.34 of the total thickness of the velocity boundary layer. The same figure for the turbulent case is about 0.04 of the thickness. Experimentally we have the maximum at about 0.2. With an estimated temperature difference to the wall of 0.2°C one gets a local Gr-number around 5·10⁹, which indicates a turbulent boundary layer.
Figure 7.30 Temperature distribution during stand still after 16 and 32 h cooling down from 53°C.

Figure 7.31 Velocities in the boundary layer vs distance to the wall 0.7 m from the top. See fig. 7.30.
Figure 7.32 Velocities in the boundary layer vs distance to the wall 0.4 m from the top. See fig. 7.30.

Figure 7.33 Velocities in the boundary layer vs distance to the wall 0.3 m from the top. See fig. 7.30.
Figure 7.34 Normalized velocity vs normalized distance to the wall, 0.7 m from the top.

Figure 7.35 Theoretical velocity profile in a laminar boundary layer.
Figure 7.36 Theoretical velocity profile in a turbulent boundary layer.

The determination of the velocity profile in the boundary layer during charging is somewhat more difficult to do but it is still possible, at least below the gradient zone, where the disturbances from the inlet are small. Fig 7.37 and 7.38 shows the results of such measurements.

Below the gradient zone the profile is quite like the one for natural convection with laminar flow although the Gr-number is around $10^9$ indicates a turbulent flow. The mean velocity in the storage is about 0.22 mm/s and much smaller than the maximum velocity in the boundary layer. This velocity is just a bit higher than during stand still conditions with the same temperature difference to the surroundings and the thickness of the gradient zone equals the one predicted by laminar theory, eq. 3.22.

It seems natural to consider this to be essentially natural convection driven by the heat flux to the wall since $Gr/Re^2 > 1000$. The velocities are those achieved in natural convection with the mean velocity in the storage added to them.
Above the gradient zone it is not possible to detect a velocity profile with a well-defined thickness. The maximum velocity in the boundary layer is essentially determined by the heat flux to the wall, while the inside of the layer is affected by disturbances from the core.

![Graph](image.png)

**Figure 7.37** Velocity profile in the boundary layer below the gradient zone during a charging period. $\nu_{st} = 0.22 \text{ mm/s}$, $\nu_{inl} = 0.31 \text{ m/s}$, $T = 36^\circ\text{C}$. 
Figure 7.38  Velocity profile above the gradient zone \( v_{st} = 0.22 \text{ mm/s} \), \( v_{inl} = 0.31 \text{ m/s} \), \( T = 52^\circ \text{C} \).

This shows that at a charging rate of 0.4 m\(^3\)/s there is a quite high degree of turbulence in the core at this level, about 1.3 m below the inlet. This should maintain the core well mixed, at least down to this level.

### 7.3.2 Measurements in the core

These measurements are performed either in a fix point or in different points in the core by traversing the front lens of the LDA.

Fig. 7.39, 40 and 41 show the results of temperature measurements and velocity measurements in a fix point in the center of the core, and in the middle of the storage in vertical direction.

One thing to notice is that there are always disturbances in the core during the starting up phase. These give rather big fluctuations in both vertical and horizontal velocities. These velocities are damped out after a while and the velocity becomes essentially zero in horizontal direction and equals the mean velocity in the storage in vertical direction.
There will be small velocities until the gradient zone reaches the measuring volume. Then the velocities start fluctuating with bigger and bigger amplitudes, until the temperature gradient in the measuring volume is zero and they reach their maximum values. These big fluctuations are then maintained throughout the charging period and we have not been able to detect any kind of regularity in them.

This shows that the core is well mixed from the top down to this level.
Figure 7.39 Temperature vs time at the level where velocities are measured, in the middle in vertical direction. $v_{st} = 0.52 \text{ mm/s}$, $v_{inl} = 0.73 \text{ m/s}$.

Figure 7.40 Vertical velocities vs time in the center of the core where temperatures were measured in fig. 7.39.
The same type of measurements done higher up in the storage show the same tendency but with much higher amplitudes of the fluctuations, although the inlet velocity is ten times smaller here. See fig. 7.42, 43 and 44. This indicates that there is a strong reduction of the maximum velocities with the distance to the inlet even without a temperature gradient in the storage.

In order to predict the decrease of these velocities, or decrease of the turbulent kinetic energy, one has to know the lengthscale for the turbulence, which determines the dissipation rate. This we can not determine in our experiments, but it will eventually be possible with the PIV-technique. As said before this lengthscale is also dependent on the magnitude of the temperature gradient, if there is one.

The mixing that is induced from the inlet is working against the natural convection, induced at the wall. This convection has a tendency to stratify the storage and will be dominant for high Ra-numbers. [17]
Figure 7.42  Temperature vs time at the level for velocity measurements, 0.35 m below the inlet. $v_{st} = 0.52 \text{ mm/s}, v_{inl} = 0.07 \text{ m/s}$.

Figure 7.43  Vertical velocities in the center of the storage where temperatures were measured in fig. 7.42.
A complete knowledge of the vertical temperature distribution in the storage would be satisfactory to be able to describe the behavior of a water heat storage in operation, but this knowledge would require a better understanding of the interaction between these phenomenas.

![Graph showing horizontal velocities in the same point as in fig. 7.42 and 7.43.](image)

Figure 7.44 Horisontal velocities in the same point as in fig. 7.42 and 7.43.

The conditions in the core are in most cases far from stable. This makes it difficult to get a good picture of the velocity field by measurements in different points in the core. The front lens of the LDA and the photomultiplier have to be separately traversed and the photomultiplier focused between measurements in different points. This determines the time delay between the measurements to at least one minute and during this time the velocity field is probably changed.

During stand still periods is it possible to make repeatable experiments and figure 7.45 and 7.46 show the results of such measurements. The temperature profile in the storage can be seen in fig. 7.30, which shows that the storage is cooling down from a roughly homogenous temperature of about 51°C. There is no temperature gradient where the velocity measurements are performed.
Figure 7.45 Velocities in vertical direction vs distance to the wall. The distance to the top about 0.3 m. Stand still conditions with $T = 51^\circ\text{C}$.

Figure 7.46 Velocities in vertical direction vs distance to the wall. The distance to the top about 0.4 m. Stand still conditions with $T = 51^\circ\text{C}$. 
The velocity in the boundary layer is roughly the same at the two levels. Theory predicts an increasing velocity deeper down in the storage. Here we can see a reversed picture, but the difference lies within the inaccuracy of the measurements. Positive velocities are downwards in the storage and we can see that inside the boundary layer there are velocities upwards. This is also what one has to expect if the continuity shall be fulfilled.

These measurements were done with a front lens of 310 mm focal length and therefore it is not possible to measure the velocities exactly at the center of the storage, that is at a distance of 400 mm from the wall.

At a level of 0.4 m below the top we detected rather high velocities upwards in the middle of the storage. Also at 0.3 m below the top one can see that the magnitude of the velocities are greater in the middle but with no dominant direction.

![Graph](image_url)

**Figure 7.47** Velocities in vertical direction vs distance to the wall. Distance to the top about 0.3 m. $v_{st} = 0.084$ mm/s, $v_{inl} = 0.012$ m/s, $T = 70^\circ$C.

These higher velocities are detected straight under the inlet and could be induced therefrom even if there is no circulation through the storage. It could be the result of a change of flow direction.
also be so that the water movements in the core, that are demanded by continuity to compensate for the downward velocities in the boundary layer, are not equally shared over the entire core but instead compensated by a strong upward stream in the center.

Results from the same type of measurements during a charging period with a small charging rate are shown in fig. 7.47. The velocities in the boundary layer are here higher due to a larger temperature difference to the surroundings. The velocities in the core are relatively small because of the low charging rate and inlet velocity. The low mean velocity in the storage lies within the inaccuracy of the measured velocities. There is a weak tendency towards higher velocities in the center.

To get a better knowledge of this, one has to get an instantaneous picture of velocities in many points in the core, for example with PIV-technique.
8. CONCLUSIONS

The main task for a heat storage in an application is to store maximum amount of heat in the available volume and when the heat is demanded it shall be delivered at a useful temperature. That is, we require highest possible energy and exergy efficiencies.

These efficiencies are clearly effected by such properties of the storage as

- inlet geometry
- insulation
- height to diameter ratio
- storing medium
- materials in the walls
- etc.

They are furthermore effected by properties of the environment where the storage will operate such as

- flowrates, determined by the available time for charging and discharging
- temperature levels
- stability of the charging and discharging temperatures
- outdoor temperature, at least for the big storages
- the amount of stored heat, is the storage completely filled or half full
- etc.

The total cycle for a storage has normally three parts namely the charging period, a stand still period followed by the discharge period. To optimize the storage is then merely to optimize its usefulness in its environment and at the end it will be to optimize the economic gain of the storage.

Our work has mainly been dealing with the storage itself and the way it will react on separate outer conditions. Which are the physical phenomena that have a major effect on the behavior of the storage and how do they interact to give the total properties of the storage when installed in an environment.

Looked upon from the outside it is enough to know the vertical temperature distribution at every time to be able to predict the behavior of the storage in an application. This distribution is however strongly affected by the
velocity field in the storage which is composed of forced convection, induced from the inlets and natural convection driven by the heat transport between the water and the walls.

Therefore we have studied both temperatures and velocities in the storage with many temperature gauges and with LDA-technique.

The final goal is of course a model for water heat storages which covers the important phenomenas under different conditions and still is simple enough to be a useful tool for the constructing engineer.

We will here list some conclusions on the influence from the different phenomenas that we have studied.

The mixing at the inlets is of major importance for the thermocline in the storage. To get a good stratification this mixing shall of course be as small as possible. This means that the stabilizing effect, the temperature difference, shall be as high as possible while the destabilizing, the inertia of the incoming water, must be as small as possible and with the water spread in horizontal direction.

Therefore the charging temperature should be high, near 100°C, for an unpressurized storage. The minimum temperature should be as low as possible, that is the temperature of the water returning from the network should be low. Furthermore these temperatures should be maintained stable since varying temperatures would give increased mixing.

The initial mixing volume is related to the Richardson-number. Experiments indicate a weak dependence for \( \text{Ri} > 0.1-0.2 \) and that the mixing volume is propotional to \( 1/\sqrt{\text{Ri}} \) for smaller values of the \( \text{Ri} \)-number. The correct choise of the characteristic length and velocity scales is not yet clearly stated and demands more work.

With the same construction of the inlets at top and bottom the behavior will be symmetric during the initial phases of the charging and discharging period and it is possible to determine the mixing volume in the same way in both cases.
The Ri-number cannot be used to calculate the mixing volume when the gradient zone moves further away from the inlet since viscous forces and thereby the Re-number will then become more and more important as the buoyancy forces decreases.

Our experiments clearly indicates that we get turbulence in the core above the gradient zone during charging. The energy in this turbulence, the turbulent kinetic energy, is decreasing with the distance to the inlets. In order to determine the dissipation rate for the turbulent kinetic energy, one has to know the relevant lengthscale for the turbulence. This is not known and is furthermore certainly effected by eventual temperature differences.

Neither is the true starting value for the turbulent kinetic energy known, when the water starts penetrating into the core. The kinetic energy is known when the water enters the storage, but then the water is spread over the cross section of the storage before it starts penetrating and this gives a decrease of kinetic energy from continuity reasons. These problems will require more work and hopefully will PIV-technique enable us to get some more information on this.

When the gradient zone moves away from the inlet natural convection at the wall and thermal diffusion across the gradient zone becomes important. Both these phenomena are active during charging, discharging and during stand still periods.

The velocities in the boundary layer are roughly proportional to the square root of the temperature difference across the wall and only to a small extent dependent on the available height and the mean velocity in the storage.

In our experiments the value of the Gr-number is of the magnitude of about $10^6$ to $10^{10}$ indicating that we can get laminar flow, turbulent flow or be in the region for transition to turbulent flow in the boundary layer. In larger storages the flow will surely be turbulent, since the significant length in the Gr-number is much bigger and the temperature difference is of the same order of magnitude.

In our storage we get a value for the quotient Gr/Re² of more than $10^3$ if Re is based on the diameter of the storage and more than $10^6$ if it is based on a boundary layer thickness. Eq. 3.19 then gives that the flow in the boundary
layer is of type natural convection. For a big storage this quotient would be of roughly the same magnitude based on the boundary layer thickness, which seems to be more reasonable to use than the diameter.

A calculation of the Re-number based on the diameter of the storage gives the result that while we have a laminar mean flow, at least far away from the inlet, the flow would be turbulent in the whole storage in a big one. If the Re-number is based on the thickness of a boundary layer it would indicate roughly the same type of mean flow in storages of different sizes which presumably is more correct.

The velocities in the downstream at the walls are of the magnitude $10^{-3}$ to $10^{-2}$ m/s and the thickness is of the order $10^{-2}$ m. Some part of this water continues down to an eventual gradient zone where it is redirected towards the center of the storage and gives a broadening of this zone. The rest will get into the core above the gradient zone and lead to a decrease of its temperature. For high Ra-numbers it will also give a stratification here. How this downstream flow is shared between the gradient zone and the core above it will demand further investigations.

Thermal diffusion will always be present but in parts of the storage where there is convection this convection will be responsible for the main transport of heat. The diffusion is of importance in the lower part of the gradient zone where the velocities are small and of course also in the immediate vicinity of the walls.

One problem that lately has been discussed in this field is whether the heat diffusion in vertical direction in the walls has a major effect on the stratification. If there would be a strong transport of heat in the steel walls this would then induce an upward velocity somewhere in the boundary layer, which we have not detected in any of our experiments. This might be due to the fact that in the area where we measure the velocities the walls are made from glass, which have a thermal diffusivity that is about 30 times smaller than that for steel.

During stand still conditions Benard convection will be a dominant phenomena, responsible for much of the mixing above the gradient zone and results in a decrease of the mean temperature above the gradient zone and eventually in a decrease of the thickness of this zone. In storages with a
steam cushion at the top there will be no Benard convection since heat is delivered from the steam to the top of the water volume.

Natural convection that is driven by heat losses through the walls and the top can be decreased by increasing the thickness of the insulation, while the part of the convection that depends on the heat stored in the steel walls only to a small extent will be effected by the degree of insulation.

The heat exchange across the gradient zone by thermal diffusion can be decreased by decreasing the horizontal area of the storage. Since the velocities in the natural convection at the walls seems to be weakly dependent on the height of the wall it must be preferable to have a high height to diameter ratio, say H/D 3 to 4.

High values for H/D will increase the influence, from natural convection at the wall, on the degeneration of the thermocline. Low values will increase the influence from thermal diffusion across the gradient zone and from Benard convection at the top.

The horizontal equilization of temperatures is so effective that the temperature field can be considered as one-dimensional, except in the boundary layer at the wall.

Suggestions for further work

There are a lot of phenomenas that should be studied in a small, well equipped model storage with both LDA-technique and PIV-technique, for example.

- The interaction between the boundary layer, the gradient zone and the core above the gradient zone. PIV-technique.

- The turbulence in the core and the gradient zone. PIV and LDA.

- The velocities in the vicinity of the inlet. LDA and PIV.

- The influence from different inlet configurations. PIV and LDA.
- The flow that is induced in and near the gradient zone at a wall with a high thermal diffusivity. LDA and PIV.

A storage of this type is now available and will be used in such investigations.

Numerical methods should be possible to use to study single phenomenas in the storage, for example the mixing at the inlet.

Based upon already done and the listed new investigations it should be possible to set up a model for a water heat storage that accounts for the phenomenas which are mainly determining the behavior of a storage under different conditions.

With a model for the storages behavior available the natural step would be to look at different applications, for example short term water heat storages in combined heat and power plants.
ACKNOWLEDGEMENTS

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REFERENCES


10. Fritsche V. (1985) Measuring program to investigate the thermal behavior of a 2000 m³ heat storage tank. BMFT FB-03E0441-A. University of Stuttgart, BRD.


THE ESTABLISHING OF A GRADIENT ZONE AND ITS EFFECT ON
THE VELOCITY FIELD IN A WATER HEAT STORAGE

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ABSTRACT

The performance of a short term water heat storage is highly dependent on the degree of temperature stratification. Important phenomena as forced and natural convection, buoyancy effects and turbulent mixing and their influence on the stratification are discussed. Results from temperature and LDA measurements are presented.

KEYWORDS

Heat storage, stratification, temperature measurements, velocity measurements.
INTRODUCTION

It has been stated by many scientists that thermal stratification is necessary to get the best performance of a heat storage in solar applications. Our experiments confirm results from the literature with the conclusions that stable stratification is possible to achieve if one follow some simple rules in designing the storage.

- The hot water inlet/outlet should be as near the top as possible and cold water inlet/outlet near the bottom.
- Loading temperature should be high and uniform.
- As low temperature as possible of the cold water.
- The inlets should spread the water horisontally with lowest possible velocity.
- Good insulation.
- Suitable height to diameter ratio.

Some of the most important phenomena that tends to disturbe the stratification are

- Turbulent mixing of hot and cold water due to inertia of the incoming water, most important near the inlets.
- Diffusion of heat across the gradient zone between hot and cold water.
- Natural convection caused by heat losses at the walls.
- Bernard convection caused by heat losses at the top, important during stand still periods.

The aim of the ongoing project is to investigate these phenomena by measuring the temperature and velocity fields in a stratified storage for different conditions. The results will be used in the modelling of such storages.
EXPERIMENTAL SET-UP

The measurements were made in a laboratory pilot plant where the storage was installed. A computer based system was used to control valves, pumps and boilers in order to get well defined conditions for temperatures and water flows.

The vertical storage has a total volume of 1.2 m$^3$. The diameter is 0.8 m and the height 2.4 m. The top inlet has an annular slot which is adjustable in order to get different horizontal velocities. The storage has twelve windows to make visualization and velocity measurements possible.

The temperature measurements were made with fortynine thermo-couples (Copper-Constantan) each one individually calibrated. The gauges are placed with 0.1 m distance in vertical direction connected to a computer based measuring converter system.

The velocities were measured with a two-component Laser-Doppler-Anemometer, LDA (Helium-Neon, 15 mW, 6328 Å) (figure 1) at two levels, in the middle of the vessel and near the top inlet. Two different lenses were used, one with the focal length 310 mm for measurements in the centre of the storage and the other one with focal length 80 mm for measurements in the boundary layer near the wall. At each point the velocities in vertical and tangential direction were measured. White-Light-Speckle-technique (figure 2) will be used to measure the third velocity component, in radial direction. This technique is a valuable complement to LDA as it gives instantaneous information of the velocities in one plane.
EXPERIMENTAL RESULTS AND DISCUSSION

The variation of temperatures during a loading cycle can be seen in figure 3 and 4. Figure 3 shows a normal loading with stable conditions and figure 4 a case with more mixing of hot and cold water.

Figure 3a and 4a shows the temperature for 8 gauges at different levels from top to bottom in the storage. In figure 3b and 4b one can see the vertical distribution of temperatures at the time when the gradient zone passes the 8 gauges.

The first part of the loading cycle is very important in establishing the thermocline in the storage. The hot incoming water will then meet the colder water in the storage and due to its inertia it will start penetrating down the storage. This is done against the buoyancy forces caused by the difference in density between hot and cold water. The penetrating depth is determined by the level the hot water will reach before its kinetic energy is converted into work against buoyancy and viscous forces.

At this level the lower part of the gradient zone will develop and this zone will then move downward with the mean velocity in the storage. From that on we get complete mixing of the incoming water into the volume above this level.

If this mixed volume is big, its temperature will grow slowly and give rise to a weak temperature gradient, see figure 4. As the gradient zone moves downwards the mixed volume will grow and thereby cause a slower growth of the temperature. This can be seen in figure 3 and 4 which show a big difference between the upper and the lower part of the gradient zone. The biggest temperature gradient is in the lower part of the gradient zone which is first established.

The gradient zone has a great influence on the velocities in the storage. Figure 5 shows that above the gradient zone one has great velocity variations in the center of the storage. These are strongly reduced by the gradient zone so that below it one find small variations around the mean velocity in the storage. This shows that one get a good mixing of the water above the zone, at least down to this level 1.2 m below the inlet. Buoyancy forces acting upwards will stop the hot water from penetrating the gradient zone.
The variation of the maximum velocity in the boundary layer near the wall can be seen in figure 6. The velocities are very high compared to those in the center of the vessel due to natural convection caused by the heat losses. But also here one get a strong reduction of the velocities when the gradient zone passes the measuring volume. The temperature difference to the air is doubled when the gradient zone has passed. Also the velocities are doubled and seem to be proportional to the temperature difference. The influence from the available height is small.

The initial penetration depth is determined by the velocity of the incoming water and of the difference in temperature between the incoming water and the water in the storage.

A modified Richardson number is defined as

\[ Ri = \frac{\Delta \rho \cdot g \cdot h}{\rho \cdot u^2} \]

where \( \Delta \rho \) = difference in density between cold and hot water
\( \rho \) = density of the incoming water
\( g \) = acceleration of gravity
\( h \) = characteristic length for the problem
\( u \) = characteristic velocity for the problem

There is a problem to define the characteristic quantities. We have used the unit length for \( h \) and the inlet velocity for \( u \).

The relation between the Ri-number and the initial penetration depth or initially mixed volume is shown in figure 7. The mixed volume is almost independent of the Ri-number if it is greater than 0.25. For smaller Ri-values the experimental results indicate that the mixed volume is proportional to \( 1/\sqrt{Ri} \).

A common problem in solar applications is the varying temperatures from the solar collectors. This means that one get cold water above hot water which gives an unstable situation with negative Ri-numbers. Complete mixing at least down to the gradient zone will occur which lowers the temperature of the available heat. The same problem will arise if one want to
produce tap water and therefore extract heat with a heat exchanger in the top of the storage.
ACKNOWLEDGEMENT

This project is financially supported by the Swedish Board for Technical Development (STU). Sincere thanks are due to Professor A. Haag for proposing the project and for guidance.
REFERENCES


EXPERIMENTAL SET UP LASER DOPPLER ANEMOMETRY

- Optical unit
- Beam expander
- Lens
- Storage
- Photo multiplier

He-Ne Laser 15 mW, 6328 Å

30-50 MHz

Tracker

60-80 MHz frequency shifter

Filter

Computer

Figure 1. Experimental set-up for LDA measurement.

EXPERIMENTAL SET UP WHITE LIGHT SPECKLE MEASUREMENT

Particles:
- Pliolite (Union Carbide)
- Diameter 30 um - 70 um
- Density 1000 kgm⁻³
- Velocity max 5 cms⁻¹

Storage

Windows

Lightsheet

Flash slot

Camera

Velocity field to be double exposed

Figure 2. Experimental set-up WLS-measurement.
Figure 3. Temperatures at different levels in the storage. Mean flow 0.6 m$^3$/h. Inlet-velocity 0.014 m/s. Ri-number 22.
Figure 4. Temperatures at different levels in the storage. Mean flow 1.4 m$^3$/h. Inlet-velocity 1.07 m/s. Ri-number 0.013.
Figure 5. Temperature, vertical and horizontal velocity in the center of the storage. Mean flow 1.07 m$^3$/h. Inlet-velocity 0.084 m/s. Ri-number 8.6.
Figure 6. Temperature in the center of the storage and maximum velocity in the boundary layer at the same level. Mean flow 1.0 m$^3$/h. Inlet-velocity 0.84 m/s. Ri-number 0.086.
Figure 7. Normalized initially mixed volume vs ln(Ri). $V = \text{mixed volume}$, $V'' = \text{total volume}$. Mean flow 0.15-1.4 m$^3$/h. Inlet-velocity 0.0035-1.15 m/s. Temp.diff. 0.9-30°C.
TEMPERATURES AND VELOCITIES NEAR THE GRADIENT ZONE IN A SHORT TERM WATER HEAT STORAGE.

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ABSTRACT

One of the phenomena that determines the degree of thermal stratification in a short term water heat storage is the exchange of water between the boundary layer at the wall and the center of the storage due to natural convection. In the present paper results from velocity field measurements using Particle-Image-Velocimetry (PIV) are presented. The experiments are performed in a small heat storage. The velocity field near the wall is measured and indicates the presence of large scale velocity cells above and below the gradient zone. The vertical extension of these cells varies during a loading cycle and is determined by the position of the gradient zone.

1. INTRODUCTION

The use of heat storages has been fairly common during the last decades and they will surely be even more used in the future. There is still no complete knowledge of all phenomena that should be accounted for in an accurate model to describe the thermal behavior. The degree of thermal stratification is very much determined by the velocity field in the storage which implies the need for velocity measurements. Earlier experiments show that the mixing at the inlet has a dominant influence on the establishing of the gradient zone. [1]

When this zone moves downwards the influence from natural convection and heat diffusion becomes more important. The earlier measurements of velocities in and near the boundary layer were performed with Laser-Doppler-Anemometry-technique which is a point method and gave the velocities in vertical and tangential direction in the storage [2].

To be able to predict the influence of natural convection on the stratification, one has to know the velocities in radial direction which determines the interchange of water between the boundary layer and the core of the storage. The Particle-Image-Velocimetry-technique (PIV) is employed here to provide information about the radial and vertical velocity components.

2. METHOD AND EXPERIMENTAL SETUP

PIV is a non-intrusive technique for the measurement of an instantaneous two-dimensional velocity-field in a fluid [3,4,5]. The fluid is seeded with small neutrally buoyant particles, Pliolite \( \phi < 36 \mu m \). A sheet of laser-light illuminates the region of interest and a multiple exposure photograph of the sheet captures the velocity field (fig 1). By shining a laser beam through the developed negative an interference fringe pattern is obtained on a screen (Young's fringes). The distance between the fringes is proportional to the velocity and the orientation of the fringes gives the flow direction in the area covered by the laser beam. The picture on the
screen is fed into a computer by a CCD-camera for an automatic evaluation and by scanning the laser beam across the negative, the velocity field is obtained. A thorough account for the data processing employed here has been given in [6].

The heat storage model consists of a 250x400x420 mm\(^3\) glass-walled tank with inlet at the top and outlet at the bottom. The flow rate was manually operated by an accurate control valve. A temperature control unit maintained the temperature of the inlet water constant within ± 0.2\(^\circ\)C. The vertical temperature distribution during a loading cycle was measured with thermocouples connected to a computer. The walls were insulated except at the light sheet entrance and at the area needed for photography.

Light from a 4 W Ar-laser (L) passed a shutter (S) and a cylindrical lens (CI) which spread the light into a light sheet (fig 1). A camera (C) (Hasselblad), loaded with Kodac Technical Pan film, was focused on the light sheet. The camera shutter was open and the exposure determined by the shutter (S). The number of exposures and time delay between them was chosen according to the velocity range to analyze.

![Fig 1. Experimental set up.](image)

3. RESULTS AND DISCUSSION

We will here discuss a sequence of pictures that shows the velocity field in and near the boundary layer at the wall of the storage. All the measurements were done at the same height in the storage during a whole loading cycle which means that the fields below, in and above the gradient zone were recorded. The inlet temperature was 43.7\(^\circ\)C, starting temperature for the storage 24.3\(^\circ\)C and ambient temperature 23.8\(^\circ\)C. The loading rate was 24 l/h giving a mean velocity in the storage of 67 \(\mu\)m/s. Fig 3 shows the temperature distribution when the individual photographs were taken.

Fig 2a shows a triple exposed photo below the gradient zone and fig 2b the corresponding evaluated velocity field. The observed flow direction towards the wall can partly be explained from continuity reasons. The natural convection at the wall will cause a slower upward motion relative to the downward mean stream, in the center of the storage. This direction of the velocities near the boundary layer indicates the presence of a large scale velocity cell, limited in vertical direction by the gradient zone and with velocities directed away from the wall further down in the storage (not seen on the picture).
The picture also shows increasing maximum velocity in the boundary layer, from 350 μm/s in the upper part to 500 μm/s in the lower part of the picture, that is on a vertical distance of 17 mm. This is what could be expected since theory predicts an increasing velocity with increasing available height, in this case the distance from the measuring point up to the gradient zone.

When the lower part of the gradient zone reaches the measuring area the velocities in the boundary layer are clearly smaller (at least below 200 μm/s) and the velocities near this layer changes so that from being directed towards the wall they become parallel to the wall and higher up in the gradient zone directed away from the wall (fig 4). This indicates another cell above the gradient zone with its lower part in the gradient zone and with a possible vertical extension up to the top of the storage.

Fig 5 shows that when water in the inner part of this layer comes in contact with colder water in the gradient zone it will penetrate a bit downwards into this water. This will cause a buoyancy force directed upwards which gives an upward velocity, at least relative to the mean velocity. Higher up in the zone (fig 6), with higher velocities in the boundary layer, the momentum of the water gives a comparably deep penetration. This causes an upward velocity that is larger than the mean velocity in the storage and will be detected as an absolute upward velocity.

Above the gradient zone even higher velocities can be detected in the boundary layer (up to 3000 μm/s) depending on a greater temperature difference to the surroundings and on a larger available height. When the gradient zone is near the measuring area the flow in the vicinity of the boundary layer is directed towards the center of the storage. When the zone moves further down the flow is instead directed towards the wall which also implies the presence of large scale convection cells above the gradient zone.

From the experiments performed one can conclude that there are large scale convection cells above and below the gradient zone. The vertical extension is determined by the position of the gradient zone and varies during a loading cycle. There is a continuous horizontal transport of water between the boundary layer and the core over the total height of these cells.

REFERENCES


Fig 2a Triple exposed photograph of the flow below the gradient zone, in and near the boundary layer. Time between exposures: $\Delta t = 1.5$ s.

Fig 2b Velocity field below the gradient zone, in and near the boundary layer. Dots represents measuring points with velocity information.
Fig 3  Temperature distribution in the storage when the actual photos were taken.

Fig 4a  Triple exposed photograph in the upper part of the gradient zone. Δt = 1.5 s.

Fig 4b  Corresponding velocity field.
Fig 5a  Triple exposed photograph in the upper part of the gradient zone. \( \Delta t = 4.5 \) s.

Fig 5b  Corresponding velocity field.

Fig 6a  Triple exposed photograph in the top of the gradient zone. \( \Delta t = 4.5 \) s.

Fig 6b  Corresponding velocity field.
PHENOMENA THAT EFFECT THE THERMAL STRATIFICATION IN WATER HEAT STORAGES.

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Introduction

It is a well known fact that water heat storages shall be thermally stratified in order to give high energy efficiencies. The benefits of stratification is even more evident if one look at the exergy efficiency, that is giving a higher value to heat at a high temperature.

We performed our experiments in a cylindrical vessel of volume 1.2 m$^3$, diameter 0.8 m and height 2.4 m. The inlets were designed as annular slots at the top and the bottom. Temperature measurements showed no significant temperature difference in horizontal direction in the storage. Thus the vertical distribution of temperatures determines the degree of stratification.

Since the temperature distribution is very much determined by the velocity field in the storage we did a lot of measurements of velocities with LDA-technique. These measurements gave the velocities in vertical and horizontal direction in the storage.

In order to measure velocities in radial direction, which for example determines the exchange of water between the boundary layer at the wall and the core of the storage, we are now using Particle-Image-Velocimetry-technique. This enables us to get an instantaneous two dimensional picture of the velocities in the storage.

We will here discuss some of the most important phenomenas that effect the thermal stratification, starting from the beginning of a charging cycle. The most important phenomena in the establishing of the gradient zone is the mixing that occurs at the inlet in the beginning of a charging cycle.

As the gradient zone moves further away from the inlet, other phenomena will effect the growth of this zone. One is natural convection at the wall, or other surfaces with temperatures that differs from the water in the storage. Another is thermal diffusion in vertical direction in the water and heat conduction in vertical direction in the walls.

Heat losses at the top of the storage will cause Benard convection during stand still periods.
Mixing at the inlet

The tendency to mix the incoming water with the water in the storage is due to the momentum of the incoming water. The stabilizing factor is the temperature difference between incoming water and water in the storage. The depth of penetration of the incoming water, or the mixing volume, is shown to be very much related to a modified Richardson-number defined as

\[ Ri = \frac{g \beta \Delta t h}{v^2} \]

where
- \( \beta \) = coefficient of volumetric expansivity.
- \( \Delta t \) = temperature difference
- \( h \) = characteristic length for the mixing
- \( v \) = velocity of incoming water

For Ri-numbers smaller than about 0.25 the mixing volume is shown to be approximately proportional to \( 1/\sqrt{Ri} \) and for larger Ri the mixing volume is only weakly dependent on the Ri-number (fig 1).

The temperature gradient in the gradient zone is very much determined by the initial mixing of the first water that comes into the discharged storage. Large mixing volumes result in weak temperature gradients and thereby a thick gradient zone. As the gradient zone moves downwards the temperature difference decreases and the mixing volume increases. The result can be seen in fig 2 which indicates a weaker temperature gradient in the upper part of the gradient zone.

In a model that should describe this mixing at the inlet one has to account for these inertia and buoyancy effects. Then it is necessary to be able to predict a correct value for the length scale and for velocities. The value of \( h \) should reflect the height over which the temperature difference \( \Delta t \) occurs. Since the gradient zone is not established this height is not known and one has to choose another adequate value.

One possibility is to define \( h \) as the vertical distance between the inlet and the outlet (used for example by Cole et.al). This would give different values for Ri for storages with the same diameter but with different heights and thereby implying different mixing volumes for those storages. This is not what could be expected from the physics for the problem unless the Ri-number is so small that the storage would be fully mixed. Instead the penetration depth and mixing volume should be the same for storages that differ only in distance between inlet and outlet. In our calculations we have used the unit height 1 m for \( h \).

There is also a problem in determining the correct value for the velocity \( v \) in the Ri-number. This velocity should reflect the velocity of the water when it starts penetrating into the water of different temperature. In most storages the water is spread in radial direction in the top and bottom of the
storage which means that the velocity decreases from continuity reasons. If one uses the inlet velocity for $v$ this would imply the same penetration depth for storages that differs in the diameter of the storage.

One could expect that it is more correct to say that the mixing volume is the same for those storages and hereby giving different penetration depth for different diameters. However this is not to expect from physical reasons since smaller diameters would imply a greater value for the height $h$ and hereby giving greater bouyancy effects. So far we have used the inlet velocity for $v$.

Our coming work will be to try to get a better knowledge of the physics of the problem, by doing measurements with LDA and PIV techniques in order to get the velocity field in the vicinity of the inlet.

**Natural convection**

When the gradient zone moves further away from the inlet the influence on the zone from the inlet will become weaker. Then the natural convection at the walls will be of importance for the further development of the zone. This is caused mainly by heat losses through the walls but also by the heat transport between water and the material in the walls in order to equalize their temperatures. The heat stored in the walls can be of great importance in a storage with a small cross section. In fact it gives rise to a significant decrease in the temperature gradient in small model storages. In full scale modells this effect will be of less importance.

Our measurements of velocities in the boundary layer at the walls indicate that the influence from the available height is rather weak. The velocities in the boundary layer reaches their maximum value over a rather small height and then remains almost constant downwards in the storage. The maximum velocity is here roughly proportional to $\sqrt{\Delta t}$ where $\Delta t$ is the temperature difference to the surroundings (fig. 3, 5, 6).

In fig 5,6 we can see that it is only when the gradient zone is just a few centimeters above the measuring volume that the velocities are to some extent effected by the available height. When the gradient zone gets further down the velocity increases up to a certain value, where it becomes constant.

The interpretation of this could be that just below the gradient zone the flow is laminar in the boundary layer. Further down the flow becomes turbulent and the velocity reaches its maximum value. The input of energy in the boundary layer is mainly used to maintain the turbulence.

The strongest influence on the velocities has the temperature gradient, since the velocities becomes essentially zero when the temperature gradient has its biggest value, and the velocity does not reach its maximum value until the temperature gradient is zero.
It seems to be so that almost no water in the boundary layer above the
gradient zone passes the zone downwards. Instead the direction of the water
is changed near the gradient zone and the water flows in radial direction
into the core of the storage. Fig 7,8,9 show results from PIV-measurements.
The triple exposed photo and the corresponding evaluated velocity field
show the situation below the gradient zone (fig 7). The observed flow
direction towards the wall can partly be explained from continuity reasons.

The natural convection at the wall will cause a slower upward motion
relative to the downward meanstream in the center of the storage. The
direction of the velocities near the boundary layer indicate the presence of a
large scale velocity cell limited in vertical direction by the gradient zone.

The pictures (fig 7) also show increasing maximum velocity in the boundary
layer due to an increasing available height, in this case the distance from
the measuring point up to the gradient zone.
It is also possible to evaluate the boundary layer thickness from the velocity
profile

When the lower part of the gradient zone reaches the measuring volume the
velocities in the boundary layer are clearly smaller and the velocities inside this
layer changes so that from being directed towards the wall they become parallell
to the wall and higher up in the gradient zone directed away from the wall. (Fig
8,9) This indicates another cell above the gradient zone with its lower part in the
gradient zone and with a possible vertical extension up to the top of the storage.

In the upper parts of the gradient zone one detects even higher velocities in the
boundary layer. Fig 9 shows that when water in the inner parts of this layer
comes in contact with colder water in the core it will penetrate a bit downwards
into this water. This will cause a buoyancy force directed upwards which gives
an upward velocity, at least relative to the mean velocity.

Above the gradient zone one detects higher velocities in the boundary layer
depending on a greater temperature difference to the surroundings and on a
larger available height. When the gradient zone is just below the measuring
volume the velocities in the vicinity of the boundary layer are directed towards
the center of the storage. When the zone moves futher down these velocities
are instead directed towards the wall which also implies the presence of large
scale convection cells above the gradient zone.

From the experiments performed in this storage one can conclude that there are
large scale convection cells above and below the gradient zone. The vertical ex­
tension is determined by the position of the gradient zone and varies during a
loading cycle. There is a continous horizontal transport of water between the
boundary layer and the core over the total height of these cells.

During stand still conditions Benard convection will be a dominant
phenomena, responsible for much of the mixing above the gradient zone
and results in a decrease of the mean temperature above the gradient zone.
and eventually in a decrease of the thickness of this zone. In storages with a steam cushion at the top there will be no Benard convection (fig4).

Natural convection that is driven by heat losses through the walls and the top can be decreased by increasing the thickness of the insulation, while the part of the convection that depends on the heat stored in the steel walls only to a small extent will be effected by the degree of insulation.

**Heat diffusion and conduction**

Thermal diffusion will always be present but in parts of the storage where also convection is present it will be responsible for the main transport of heat. The diffusion is of importance in the lower part of the gradient zone where the velocities are small and of course also in the immediate vicinity of the walls. Fig 2 shows that the measured temperatures in the gradient zone differs from the theoretical ones where only thermal diffusion is taken into account. This difference is due to heat transport caused by forced and natural convection.

One problem often discussed is whether the heat conduction in the walls has a major effect on the stratification. If there would be a strong transport of heat in the steel walls this would induce an upward velocity somewhere in the boundary layer, which we have not detected in any of our experiments. This might be due to the fact that in the area where we measure the velocities the walls are made from glass, which has a thermal diffusivity that is about 30 times smaller than that for steel.

**Conclusions**

The mixing at the inlet could be reduced by keeping the inlet velocities low and the temperature differences high. The water at the inlet is normally spread horizontally which means that they will decrease from continuity reasons. A large storage diameter would then give small velocities and therefore small penetration depths. A small diameter causes higher velocities and larger penetration depths which means larger work against buoyancy forces. Its difficult to choose the best height to diameter ratio (H/D) that would give the smallest mixing volume, which is relevant for the efficiency of the storage.

The heat exchange across the gradient zone by thermal diffusion can be decreased by decreasing the horizontal area of the storage. Since the velocities in the boundary layer at the walls seems to be weakly dependent on the height of the wall it must be preferable to have a high H/D ratio, say H/D.

High values for H/D will increase the influence, from natural convection at the wall, on the degeneration of the thermocline. Low values will increase the influence from thermal diffusion across the gradient zone and from Benard convection at the top.
The horizontal equilization of temperatures is so effective that the temperature field can be considered as one-dimensional, except in the boundary layer at the wall.

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REFERENCES


Fig. 1 Initially mixed volume vs natural logarithm of Ri-number.

Fig. 2 Normalized temperatures vs normalized time at three levels in the storage. Dots for experimental values and lines for theoretical results. t'' is the time for the charging of one storage volume.
Fig 3. Maximum velocity in the boundary layer vs square root of the temperature difference.

Fig 4. Temperature vs height at different times during a stand still period after 0, 5, 15, and 21 h.
Fig 5. Temperature vs time at the inlet level where the velocities are measured. $v_{st} = 0.22 \text{ mm/s}$, $v_{inl} = 0.31 \text{ m/s}$.

Fig 6. Maximum velocity in the boundary layer vs time
Fig 7a. Triple exposed photograph of the flow below the gradient zone zone, in and near the boundary layer. Time between exposures: $\Delta t = 1.5$ s. $v_{stor} = 67\mu m/s$

Fig 7b. Velocity field below the gradient zone, in and near the boundary layer. Dots represents measuring points with no velocity information.
Fig 8a. Triple exposed photograph in the upper part of the gradient zone. $\Delta t = 4.5$ s.

Fig 8b. Corresponding velocity field.

Fig 9a. Triple exposed photograph in the top of the gradient zone. $\Delta t = 4.5$ s.

Fig 9b. Corresponding velocity field.
Use of Video based Particle Image Velocimetry technique for studies of velocity fields in a water heat storage

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Abstract

A video based Particle Image Velocimetry technique has been developed. The technique is particularly suitable for measurement of small velocities, below 3 cm/s. It has proved to be useful for the documentation of non stationary velocity fields in a water heat storage. An ordinary video camera is used to record the in plane movements of particles in a light sheet in seeded water. The hardware used, the experimental method and the accuracy of the method are discussed. The use of two commercially available software's, NIH-Image and IGOR for the analyses is described. Examples of velocity fields are presented, showing that the measuring technique can be used for studies of mixing near the inlet of the storage and exchange of water between the boundary layer and the core.
1. INTRODUCTION

In most applications of heat storage where water is used as the storage medium it is essential to maintain a high degree of thermal stratification. Many investigators have shown that the overall efficiency of storage systems is improved with thermal stratification, especially in solar applications, see for instance (Han, Wu and Christenssen 1980).

Degradation of the thermal stratification is caused primarily by mixing of hot and cold water during the charging of the storage and by natural convection along the walls when heat is lost to the surroundings.

Mixing of hot and cold water due to forced convection near the inlet has been studied by the authors in a previous work based on temperature measurements in a model storage. It was concluded there that the mixing near the inlet is related to a modified Richardson number (Dahl and Hermansson 1988). It was not possible to determine the correct length and velocity scales that should be used to predict the mixing in other storage's. This requires determination of the velocity field close to the inlet.

Natural convection near the wall caused by heat losses to the surrounding has also been studied earlier by the authors (Dahl and Hermansson 1988) using two-component Laser Doppler Anemometry. Vertical and tangential velocities in the boundary layer were then recorded. The effect on the thermal stratification is however determined by the exchange of water between the boundary layer and the core. The radial velocity component, which affects this process could not be measured with this technique.

The need for more complete mapping of the flow field led to the use of the PIV-technique (PIV stands for Particle Image Velocimetry) based on multiple exposed photos of particles in seeded water. (Dahl, Hermansson, Gren, Benckert 1991). This technique gave information about the inter-change of water but only a small area near the wall could be studied, since the evaluation of the Young's fringes was very time-consuming. Another problem with this technique is that the appropriate time delay between exposures is determined by the magnitude of velocities to study and has to be set before recording. The velocity range that can be detected is quite narrow and a bad choice of the time delay will give poor results. The experiment must then be repeated.
It was therefore of interest to find another measuring technique that would make it possible to record movements in the whole storage during a complete charging cycle. The new technique must give good resolution in space and enable accurate determination of velocities in a wide range, from a few $\mu m/s$ up to the maximum velocity near the inlet in the laboratory storage, a few $cm/s$. Video based PIV where particle trace pictures are created by adding many frames from a video recording of the flow with seed particles appeared to be a step in this direction. This technique however had not been used for studies of flow in thermal storage's before. Problems with particle tracing and experimental accuracy could not be excluded, in particular since the available resources made it necessary to use an ordinary video camera and tape recorder.

The present study has two objectives, namely first to adapt the video based PIV-technique for use in studying flow fields in water heat storage's. Secondly to generate data on velocity fields, which could be used for modelling of the thermal behaviour in water heat storage's and for comparisons with numerical calculations of the flow conditions.

In this paper the emphasis has been placed on presentation and evaluation of the PIV-technique.
2. THE PARTICLE IMAGE VELOCIMETRY TECHNIQUE

2.1 Different approaches to PIV

A comprehensive description of PIV-techniques applied for velocity measurements in fluids may be found in (Hinsch 1993; Dudderer et al 1988; Adrian 1991) and a short summary in (Schmidt and Löffler 1993).

A common feature of most PIV methods is to introduce a light sheet into a fluid, were the fluid is seeded with particles that can follow the flow in an inertia-free way. The particle movements in the illuminated plane are usually recorded by making double or multiple exposures on a single photographic frame.

The analyses can be done locally, point by point, by shining a laser beam through the negative and use the Young's fringe analyses method for the velocity evaluation. It is also possible to use a global, component by component, Fourier filtering technique where the whole image is illuminated simultaneously by an expanded and collimated beam. When the aperture filter is offset from the optical axis in a given direction the filtered image appears covered with light and dark fringes. These fringes are contours of uniform velocity components. Both these methods require special attention to handle the problem with the direction ambiguity, for example by using different intensity distributions for two successive particle image patterns.

Digital PIV or DPIV is a method where a video recording of the light sheet is made instead of the photographic record of the tracer particles in the flow. Records are obtained with the frequency of the video system, and data are stored and then evaluated digitally. Schmidt and Löffler used a CCD (Charge Couple Device) camera for recording of double exposed pictures. They used image processing based on auto correlation of image sections to capture the velocity fields.

The method used in the present work is based on continuos recordings with an ordinary video camera. Commercially available software is used for image processing and evaluation of the velocity field. This technique enables recording of a complete experiment on videotape and allows interesting parts of the transient flow to be analysed afterwards. By using a frame grabber, an optional number of frames with appropriate time difference can
be captured from the video tape and stored in the computer. To visualise the flow, particle trace pictures can easily be created without ambiguity of the direction of the flow.

2.2 PIV-technique used in this study

The Particle Image Velocimetry technique used in this work is based on continuous video recordings of particle movements in seeded water. The experiments were performed in a model of a water heat storage, see section 3 for details. The particles in a vertical symmetry plane inside the heat storage are illuminated with a light sheet, created with a laser and a cylindrical lens (Fig. 1).

![Figure 1. The Particle Image Velocimetry equipment.](image)

It is possible to observe the separate particles even if the view covers the whole storage. To get good accuracy in the evaluation of velocities, the camera (Panasonic S-VHS, NV-MS1, recording 25 frames/s) is normally positioned quite near the storage. A 450 mm focal length close up lens was mounted on the video camera lens, to enable focusing on the light sheet on an area of up to 7 times 10 cm. Each close up view then covers roughly 10% of the light sheet. It is therefore necessary to change the view to record movements in the whole storage, while still maintaining high resolution.

The fact that the walls of the storage are cylindrical and the refraction index differs from that of water means that it is not possible to see the light sheet up to the wall. The camera is normally positioned on a line through the
centre of the storage, perpendicular to the light sheet. To enable observations in the boundary layer the camera has to be placed in an angle 5-10° to that line. This will decrease the area that can be focused and thereby be reproduced in a good way. The size of the area depends on the degree of zooming, the focal length of the used close up lens and the aperture. The useful part of the picture is then reduced up to 50% in horizontal direction.

It is essential for the PIV-methods in general that the particles follow the fluid in an inertia free way. The particles used in this study are grained from a material offered under the name Pliolite AC, a styrene - acrylic monomer with specified density $\rho_p = 1.03 \cdot 10^3$ kg/m$^3$. The actual density was determined by studying the buoyancy of the particles in experiments with salt solutions of different concentration. These tests indicate the density to be $\rho_p = 1.034 \cdot 10^3$ kg/m$^3$. Two different size fractions have been used, 38 to 76 μm and 20 to 36 μm. Both fractions gave good results and the smaller particles were only used for studies of natural convection, when the falling velocity is of importance. The maximum particle concentration was about 20 particles per cm$^2$ of the light sheet.

The evaluation of velocity fields is performed in four steps. First the particle movements are recorded on tape with video camera. A predetermined number of frames with a certain time delay are then read from a video recorder, Panasonic VCR, NV-FS1 LCD-Digital Scanner, into the computer (Apple Macintosh Quadra, 24 Mb primary memory) through a frame-grabber, Quick Capture, Data Translation board with a 6-bit grey scale resolution. This reading is handled by the Public Domain program NIH-Image (Rusband 1993) from the National Institute of Health Research Services Branch (NIMH). The pictures are further processed in this program as discussed below and the co-ordinates for the particles are calculated and stored on file.

Determination of velocities, to be further discussed below, is finally performed within the program IGOR (WaveMetric). Special routines for finding related particles in separate pictures as well as for calculating and displaying velocities were developed as a part of this study.
2.3 Image processing

The program utilised for image processing is the powerful NIH-Image 1.49 (Rusband 1993) version for PAL run on the computer. The frame grabber used has the maximum capacity to read 25 frames per second. If full sized pictures are captured they can be transferred to the computer memory at a rate of 15 frames per second. This rate can be increased if the size of the captured part of the image is reduced. The number of frames to read and the real time delay between frames is set from the NIH-Image menu according to the velocity range to study and the pictures are displayed on the computer screen simultaneously with the reading.

The frames are stored in the computer. The number that could be stored is determined by the size of its primary memory and the size of the pictures (400 kbytes for a full size picture). When the frames are stored in the computer memory it is possible to eliminate noise by filtering. The effect is illustrated in figure 2.

![Figure 2. Particle trajectories achieved from multiple frames, (2a) without and (2b) with noise reduction.](image)

Using NIH-Image it is possible to choose what size and brightness of particle spots that should be kept for further evaluation. The size is normally chosen between 5 and 25 pixels and the threshold level between 100 and 200 in a range of 256 to end up with an appropriate number of particle spots to analyse. The particles are not resolved on the video film and therefore the size of these spots will not show the accurate size of the particle but instead be more related to the amount of scattered light.

If a number of frames are superimposed it is possible to display particle traces, where each exposure is given its own colour or grey scale. These
traces are an effective tool for showing the flow and are created with little
time effort. Examples of pictures generated in this way are shown in figures
2, 6, 7, 8 and 9.

It is also possible to divide the pictures in separate parts for the evaluation
of velocities. This makes it possible to get more out of pictures with uneven
illumination or large velocity ranges (Fig. 7).

The Image processing program is finally used to determine the co-ordinates
for all the particle spots that are within a band of size and sufficiently
illuminated to be accepted as predictable particles. These co-ordinates are
stored on file for further analysis of velocities.

2.4 Evaluation of velocities

The magnitude and direction of velocities are calculated and displayed in
the program IGOR 1.26, a commercially available graphing and data analysis
tool from WaveMetrics. This program is suited for handling large amounts
of data and allows utilisation of user-made macros. The co-ordinate data
sets, normally 5 to 10 files, are read into IGOR and the further calculations
are performed in a number of macros developed as part of this study.

The main problem, which requires most computer time effort, is to deter-
mine which spots in all images that represent the same particle captured at
different real times. The procedure used can be described in the following
way.

Let there be n files containing the co-ordinates for the particles on n frames,
captured at consecutive and equal time steps. For each particle on the
second frame, the distance and direction to every particle on the first and
third frame is calculated. Particles in frame one and three within a certain
distance from the particle in frame two are saved for further evaluation
(Particles inside of the large circle in figure 3).
For each particle in frame two, vectors representing the movements from all particle positions in frame one to this particle in frame two are calculated. If these vectors represent possible displacements due to the maximum velocity, the vectors are added to the position for the particle in frame two, giving a possible position in frame three. Around this point is a circle with radius given by maximum change of velocity drawn (Fig. 3).

There are two criteria for selecting possible traces, first the particle in the third frame must be within this circle, second the difference in direction must be within the prescribed limits. Otherwise no particle trace has been found. If there is more than one particle within these limits, the best fitted particle due to direction and velocity is accepted for the true particle trace. A velocity vector is calculated, which represents the velocity at the position for the particle in frame two.

It is necessary to give the limits when the evaluation starts to avoid that the program spends much time on calculations of velocities, which do not represent actual particle trajectories. The range is determined by expected values for the maximum velocities, maximum change of velocities and maximum change of direction, during one time step in the problem. These limits can be determined from the superimposed pictures showing particle traces, accomplished for different time steps in NIH-Image (Fig 6a-e). These
traces are easy to achieve and if the highest velocity in the problem is not known, it is necessary to start from the smallest possible time step, 1/25 s.

This makes it possible to detect the highest velocities that are allowed by the method and by increasing this time step it will be possible to detect arbitrary small velocities. It is easy to determine displacement in Image and hereby the velocity and change of velocity.

See for example fig 6a, where the highest velocity is detected near the inlet at the upper left corner. The geometric scale in the figure is known in Image and it is therefore possible to measure the displacement during one time step and change of displacement between consecutive time steps for the particles at the inlet. This is determined with an accuracy on pixel level after zooming on this area. Maximum displacement is found to be 1.4 mm and maximum change of displacement 0.5 mm during 0.2 s. This gives the maximum velocity 7.0 mm/s and maximum change of velocity 2.5 mm/s. The limit for the maximum velocity will then be chosen to 8.0 mm/s and for maximum change of velocity to 3.0 mm/s.

The procedure is repeated through figure 6b-6e and for example in figure 6e, suitable to evaluate the small velocities in the lower part of the picture, the appropriate limits could be 300 μm/s and 100 μm/s respectively.

Still if an inappropriate choice of the limits is made, for example if the limits adopted on figure 6a was used for figure 6e, the amount of particles inside the large circle would be too many for the computer to analyse (>200). The result after several hours of computer time will be randomly spread velocity vectors. On the other hand, if the lower limits are used for figure 6a, only the smallest velocities will be detected and with poor accuracy due to small displacements. Both results will show that the main flow pattern is not captured when comparisons are made to the particle trace pictures, and a better choice should be made.

The process described above for particles in frame two, is repeated for all particles in frame three and so on up to frame n-1. This results in a number of velocity vectors, detected at certain particle positions. The particle coordinates and the velocity vectors are stored in the computer and separately displayed in graphical form (Fig. 6f, 7b, 10a).
The evaluation of velocities can be enhanced by dividing the viewed area into parts before running the IGOR program. This separation is accomplished inside NIH-Image and the areas are chosen according to one of three criteria: the area should have homogenous illumination, homogenous velocities or contain an appropriate number of particles. Most of the time for the evaluation is spent on the calculation of velocities in IGOR. Thus the number of particles in each frame of the area should be below 200 to have reasonable computation time (a few hours).

2.5 Accuracy of the method

First it should be pointed out that the use of an ordinary video camera, recording 25 frames per second, is suitable only for relatively small velocities, below 0.03 m/s. This limitation is due to the problem of identifying the same particle on many frames if it has moved more than roughly 20 particle diameters during one time step. The velocity range could be stretched by using high speed video recording, by sacrificing resolution, and to some extent by using larger and fewer particles.

The overall accuracy of this method within its limitations is determined by the inaccuracy in every step involved in the determination of the velocities. There are essentially four types of errors, which relate to deviations between flow and particle trajectories, limited resolution of the recordings, geometrical positioning and finally interpretation of particle traces. Each type of error and its effects will be discussed below.

One possible source for errors is that the particles might not follow the flow as a result of different densities of the fluid and the particles. There are two effects to consider. The particles will tend to fall in the fluid as a consequence of their higher density and may not follow the flow in accelerated motions. The latter effect is not important in the type of flow studied here. Calculations with Stoke's theory, which is valid for spherical particles and low relative velocities, indicate a terminal falling velocity of 24 µm/s for particles with diameter 35 µm and 96 µm/s for particles with diameter 70 µm, in stagnant water at 20°C. This is in good agreement with own experiments which gave a terminal velocity of 20 µm/s for the bigger particles in size fraction 20-38 µm. See appendix A for more details.
The falling motion of the particles is important only where vertical velocities are very low, as for example in the core of the storage. Calculations in appendix A show that the radius of the particle orbit will differ less than 0.05% from that of the fluids, for typical values of the main flow acceleration in a curvilinear movement. The accelerations are estimated to be below 0.1 m/s² in this storage. The density difference apparently leads to a systematic error in the vertical velocity less than 0.02 mm/s downwards, when the smaller particles are used. The error can be estimated to be of minor influence on the results everywhere in the storage except in the core, were the mean velocity is of the same magnitude as the falling velocity.

The video camera introduces some other possible sources for errors. Spatial distortion in lenses is considered to be negligible since measurements are performed in a limited area. The limited resolution in number of pixels for the covered surface is however an important reason for inaccuracy. Closer view gives better resolution in space and thereby better accuracy in the translations determined from the video frames. The camera has the specified horizontal resolution of 460 lines and vertical resolution of 625 lines.

CCD-outputs from the camera are low-pass-filtered, approximately 3 pixel for a 10-90% step and stored as an analogue video signal. As luminance from a particle is gauss distributed and the centre of gravity for the particle is used, this distortion is of minor significance. The frame grabber will give a square pixel-ratio by super sampling the video signal and no information is therefore lost.

When the frame is read to the computer, the frame grabber card digitises the video signal with a horizontal resolution of 768 pixels and vertical resolution of 512 pixels. The lowest resolution in pixels is determined by the camera horizontally and the frame grabber vertically. The maximum size of the viewed area, used for velocity measurements in the experiments, is 10 cm times 7 cm giving the resolution (460 pixel/10 cm) 46 pixel/cm horizontally and (512 pixel/7 cm) 73 pixel/cm vertically. More narrow views give better resolution.

Positioning of the camera and calibration of distances in the storage is difficult if no fixed positions are viewed. To overcome this difficulty, a grid was introduced on the front glass, giving shadows in the storage with pre-calibrated distances, accurate within 1 mm. The inaccuracy in vertical posi-
tioning of the viewed area is less than 2 mm. Distances inside the storage, and thereby also velocities are detected with an accuracy mainly determined by the resolution in pixels per unit length. Near the wall horizontal distances have to be corrected for refraction in the cylindrical Plexiglas wall.

Errors could also be due to tracing mistakes. In fact there is a higher probability for particles with low velocities to be detected many times within the light sheet. If the particle is turning around and is not perfectly spherical, it is not obvious that the centre of the spot always represents the centre of the particle. A particle could also be partly illuminated causing the centre of the light spot being departed from the centre of the particle.

The IGOR procedure for finding traces could also lead to mistakes by identifying unreasonable particle trajectories, especially if large velocity limits are chosen. However this can be avoided by doing the evaluation with different time steps (described above) and by using samples of frames captured with small delays in real time to get a more complete field information.

The limited resolution is the most important factor determining the overall accuracy of the method. The inaccuracy of determined velocities can be reduced by choosing close up views and by use of large time intervals between captured frames. Large time intervals leads to problems with tracing of particles however. An optimal time step can be achieved for each velocity range as illustrated in figure 4.

![Figure 4. Illustration of the impact from choice of time interval between captured frames on the total error due to resolution and tracing mistakes.](image)
An estimate of the best accuracy that can be achieved with the method could be as follows. The estimate is made for a close-up view of an area 3 cm times 2 cm and maximal detectable translation 20 particle diameters, here 1.2 mm, is used. Translation of particles is considered to be known within 1 pixel, since the centre of gravity is determined on sub-pixel level. Resolution will be poorest in horizontal direction as described above. Here one pixel corresponds to 0.066 mm giving the error 5.5% of the highest velocities.

Calibration of distances introduce a possible error of at most 5%. These two errors adds up to a maximum error of about 10% of maximal detected velocity for a close-up view. The falling velocity has the same absolute value independent of the magnitude of the detected velocities and the introduced error should be added to those above.

If for example the time delay is optimised to measure velocities up to 1 mm/s, the absolute error would be 0.1*1 mm/s = 0.1 mm/s when the two first sources are considered. The discrepancy due to the terminal falling velocity only affects the vertical component and can to some extent be accounted for. However, an error of at most 0.02 mm/s could be added to the result above if the smaller particles are used and 0.1 mm/s if the bigger particles are used. For a detected velocity component \( v \) the interval for the correct velocity would then be given as

\[
v-0.10 < v < v+0.10 \text{ mm/s}
\]

for horizontal velocity components and

\[
v-0.12 < v < v+0.10 \text{ for the fraction } 20-38 \mu\text{m and}
\]

\[
v-0.20 < v < v+0.10 \text{ mm/s for the fraction } 38-76 \mu\text{m}
\]

for downward velocity components.

Several possible sources for errors were eliminated by using proper procedures.

Reading of frames from video recorder to computer through the frame grabber can for instance introduce some errors. Each frame is composed of 625 horizontal lines where the odd lines are recorded during 1/50 s and the even lines during the next 1/50 s. A complete frame is recorded in 1/25 s with a time delay between consecutive lines of 1/50 s. This means that for small particle spots, covering roughly one line in vertical direction, there could be an inaccuracy of 0.02 s in real time. The reason is that particle spots
can either be recorded on odd or even lines. This is not a serious problem if close up filming is used, when particle spots cover many lines, but could be severe if small particles and wide views are used together with short time delays between frames. The smallest time delays used in this investigation is 0.2 s for close up views and the particle spots used for determining velocities cover at least 5 pixels.

Another possible source for errors is the fact that the Macintosh computers work with 60 Hz ticks whereas the video recorder and the frame grabber card works in a 50 Hz system. This error has been eliminated by introducing an option in NIH-Image to have the program running in a 50 Hz environment.

The selected time delay between captured frames has to be a multiple of 1/25 s. However lack of synchronisation between the video recorder and the computer could lead to an error of at most 1/25 s in the time delay between frames selected for evaluation. In the test reported here the synchronisation was checked by capturing pictures from a cyclic event which showed equal time delay between pictures.
3 MODEL HEAT STORAGE USED FOR EVALUATION OF THE VIDEO BASED PIV-TECHNIQUE

3.1 Requirements on the test object

The experiments have to be performed under well defined conditions. For a heat storage this means that inlet temperatures, velocities and flows as well as heat losses can be controlled and measured with acceptable accuracy. The device must allow both charging and discharging cycles.

Detection of velocity fields with PIV-technique, requires that the inside of the storage can be illuminated with a light sheet to visualise the movement of the seeded water. Therefore the storage must have transparent walls. In order to allow illumination of a whole cross section, it must not be too large and with the used equipment the maximum size is about 50 cm.

The storage should have a geometry as close to that of real storage as possible, that is cylindrical shape. Still it must be possible to make video recordings with smallest possible optical distortion of the picture. This is more difficult to achieve for the cylindrical shape.

If temperatures and velocities are to be measured only in a vertical plane axi-symmetric geometry and boundary conditions is required.

All these requirements were taken into account when the model storage was designed.

3.2 The model storage

The storage vessel used is made of Plexiglas, has cylindrical shape and a vertical axis. It has a height of 420 mm, diameter of 200 mm and wall thickness 3 mm, giving a total volume of 12.5 dm$^3$. The storage vessel is connected to a network of circulation tubes to enable different types of experiments like charging, discharging and mixing to homogenous temperatures. The set-up is shown in figure 5.
In order to minimise optical distortion, the storage is surrounded by water contained in a rectangular tank that measures 400 mm * 250 mm * 420 mm. Its total volume is 42 dm$^3$ and the thickness of the glass walls 8 mm. This gives media with the same refraction index on both sides of the cylindrical wall and provides a plane wall towards the camera. The surrounding tank also makes it possible to simulate different conditions that affect the heat losses from the storage, since the temperature level and water flow can be controlled in both water volumes.

The top inlet/outlet of the storage is made of a circular plate with diameter 100 mm. The water enters the storage from a vertical pipe at the top. To
make sure that the water is uniformly distributed in radial direction, a circular joint is mounted on the plate to provide a narrow slot for the incoming water. The size of the slot can be varied by moving the pipe and plate in vertical direction. Most of the experiments were performed with 2 mm slot, giving an inlet velocity of 5 mm/s for a flow rate of 12.5 dm³/h. This corresponds to a one hour loading cycle.

The bottom inlet/outlet is also made of a circular plate of diameter 100 mm. The width of the slot has been maintained at 5 mm during these experiments.

To get the same boundary conditions all around the storage, the temperatures have to be evened in horizontal direction on the outside of the storage. The water fed to the outside is therefore introduced at the top in horizontal direction.

3.3 Temperature measurements and control

All the heat to the system is supplied by an accurate electric primary heater, Lauda Immersion Thermostat mS/2 with 20 dm³ water store, enabling accurate temperature regulation (within 0.1°C). The cold water withdrawn from the bottom of the storage is heated in two steps, first by a heat exchanger inside the primary heater and then by another in the open tank (Fig. 5).

The water on the outside of the storage can also be heated in two steps. In this way the temperature of the water, fed to the outside of the storage, differs only a few tenths of a degree from the temperature of the water entering the storage. This is necessary since the two water volumes are separated only by the Plexiglas wall, giving a high overall heat transfer coefficient. The loading rates can be regulated to give the same mean velocity on both sides so that the gradient zones will move in parallel. This also makes it possible to perform experiments with nearly no natural convection along the walls of the storage.

The temperature sensors used are thermocouples of the Copper-Constantan type. The gauges are connected to a DataTaker, DT100 LABORATORY LOGGER, which scan all the gauges with certain time intervals. The calcu-
lated temperatures are stored on a Personal Computer. The reference junction for all the gauges is in an Aluminium block inside the DataTaker and the temperature of this block is measured with an LM335 temperature sensor.

All the gauges were separately calibrated in the range of 15 to 55°C before the experiments started. The inaccuracy is therefore considered to be less than 0.2°C during these experiments.
4. MEASURED VELOCITY FIELDS IN A STRATIFIED MODEL HEAT STORAGE

The performance of the video based PIV technique can best be illustrated by some examples of measured velocity fields. The measurements focused both on studies of the mixing that occurs near the inlet and on the exchange of water between the boundary layer at the wall and the core of the storage.

In all figures showing particle traces discussed below the movements illustrated are from light to dark grey. In all vector plots the vector scale is given in the figure and when the axes have the same scale both magnitude and direction of the resulting vectors are correct. If the scales differ one must have in mind that the magnitude of the vectors must be evaluated component by component and that the direction shown must be corrected accordingly.

4.1 Flow pattern during charging

Figure 6 shows a sequence of particle traces achieved near the inlet during charging of the storage with flow rate 200 ml/min, giving an inlet mean velocity of 5.0 mm/s. The figure shows traces, at the same position in the storage, achieved from 10 pictures separated in time in figure 6a by 0.2 s, in figure 6b by 0.5s, in figure 6c by 1s, in figure 6d by 2s and in figure 6e by 5s. The sequences are captured from the video-tape after roughly 9.5 minutes charging, which means that the top of the gradient zone is in the lower part of the picture. Inlet temperature was 24.2°C and initial temperature in the storage 21.6°C.

The particle traces shown in figure 6a-e represent a velocity field with large spatial variations. It is therefore suitable to illustrate the benefit of the possibility to chose several different time steps for evaluation of the velocity vectors. Figure 6a gives the velocity distribution very close to the inlet. Figure 6b shows the vertical flow near the wall. In figure 6c it is possible to observe how the flow near the wall partly changes direction towards the centre of the storage, caused by the buoyancy forces. Figure 6d shows the flow changing again, now to vertical direction, mainly for continuity reasons.
Figure 6. Particle trajectories achieved near the inlet after 9 min 30 sec of charging at 200 ml/min. Temperature difference 2°C between the incoming water and the water in the storage. Particles of size 38-76 μm.

a) Time separation between frames 0.2 s.

b) Time separation between frames 0.5 s.
c) Time separation between frames 1.0 s.

d) Time separation between frames 2.0 s.
e) Time separation between frames 5.0 s.

f) The resulting velocity field after evaluation of particle traces.
The time separation (5s) used in figure 6e gives particle traces only for very low velocities, while particles seem to be randomly distributed in areas with higher velocities.

The velocity field visualised above was used to evaluate velocity vectors as described earlier. The evaluation was separated in parts where different time steps was used, due to the large range of velocities in the area. The maximum velocities are about 7 mm/s and the smallest detected velocities in the area about 200 μm/s. The resulting velocity vectors are shown in figure 6f.

The flow pattern is typical for a back step and is mainly caused by separation behind the inlet edge. The interaction with the wall causes a main stream going downwards but also a horizontal stream towards the centre of the storage.

4.2 Flow pattern during discharging

Figure 7 shows results of measurements near the inlet, at the bottom of the storage, during a discharging experiment where colder water was entering at the bottom. Figure 7a shows traces achieved from 10 pictures, separated by 1.0 s. The sequences are captured from the video-tape after roughly 20 min discharging, at a flow rate of 180 ml/min. Inlet temperature was 24°C and initial temperature in the storage 39°C.

Figure 7a was divided into 6 separate parts before the calculation of velocity vectors shown in figure 7b. This was motivated by the large range of velocities, inhomogenous illumination and a high number of particles.

The velocity vectors in figure 7b show clearly that the velocity field is affected by the gradient zone. This can be seen in the left part of the picture where the vertical velocities are redirected horizontally by buoyancy forces. The water at the inlet is supplied to the storage in horizontal direction at the bottom and meets the water stream from the boundary layer. Both streams counteract and result in a vertical up-wards flow near the inlet. This vertical flow first appeared close to the wall and then gradually moved towards the inlet. This can be explained by the growth of the boundary layer when the gradient zone moves upwards.
Figure 7 a. Particle trajectories achieved near the bottom inlet during discharging. The picture shows the division into separate areas for the evaluation of velocities. Particles of size 20-38 µm were used in the experiment.
Figure 7 b. The resulting velocity field from figure 7a. The center of the storage is at -50 mm.

4.3 The boundary layer flow

The possibilities to use the present PIV-technique for studies of natural convection boundary layers can be illustrated by a test where the gradient zone in the storage was allowed to deteriorate.

The experiment was carried out after charging of the storage with water of 34.8°C. The initial temperature of the storage was 22.6°C. During the velocity measurements there was no flow into or out from the storage with the gradient zone established at mid-level. The heat losses through the wall were comparatively high.
The temperature difference between the inside and the outside of the wall was about 1.2°C when the sequences shown were recorded. The overall heat transfer coefficient is estimated to be 30 to 45 W/m²K depending on the temperature difference, in agreement with a cooling experiment performed.

The temperature distribution with temperatures shown as point values, at the levels where gauges are sited can be found in figure 8e. Since the stand still had to be preceded by a charging, in this case during 35 min to get a gradient zone in the middle of the storage, the times are given from the beginning of the charging cycle. There is a weak stratification above the gradient zone and the effect of this on the boundary layer can clearly be seen in the particle trace pictures.

Figure 8a-d show the particle traces, achieved at four separate levels above and in the gradient zone. Particles are moving from light to dark grey and in a level from top down to 4.3 cm one can see the water in the core of the storage moving upwards and towards the wall (Fig. 8a). Here water is fed into the top of boundary layer. The next picture shows traces at level 10 to 13 cm from the top (Fig. 8b). Buoyancy forces due to the weak stratification will here decrease the velocity in the inner part of the boundary layer. This flow is redirected towards the centre and the water is fed into the stream of water going upwards.

Further down the stratification is stronger and the buoyancy effect even more manifested (Fig. 8c). The boundary layer is here continuously tapped on water and the thickness of the layer becomes smaller in the lower part. Even further down the flow is stagnating, affected by the buoyancy forces.
Figure 8 a-d. Particle trajectories in the boundary layer near the wall during stand still conditions. Distances from the top of the storage is marked in the separate pictures. Temperature difference to the surrounding 1.2°C. Particles of size 20-38 μm were used in the experiment.
Figure 8 e. Temperatures in the storage when the particle traces in 8a-d were recorded.

The experiment was continued for many hours maintaining roughly the same temperature difference to the surrounding. This was achieved by circulating and cooling the water on the outside of the storage. Figure 9a-d show the boundary layer at the same level in the storage at different times. The level selected for measurements was that where the strongest temperature gradient was established during the charging. The change of the boundary layer with time is apparent.

The temperature variations for the gauges above and below the gradient zone show that the temperature gradient gets weaker and weaker and finally becomes zero (Fig. 9e). A boundary layer will establish again during this decrease of the temperature gradient and becomes stronger, the weaker the gradient gets.

From the observations of the boundary layer flow it is possible to determine the velocity distribution inside the boundary layer. When running velocity evaluations, based on particle traces as in figure 9, the number of velocity vectors from each run is not sufficient to give an accurate picture of the boundary layer flow. More information can be obtained however by taking the advantage of the video record of the flow.
Figure 9 a-d. Particle trajectories, achieved at four different real times, in the boundary layer near the wall, 17-20 cm below the top. Stand still conditions and temperature difference to the surrounding 1.1°C.
When the velocity field is comparably stable as in the boundary layer, it is possible to evaluate many series of pictures centred around the same real time. In this way a number of velocity plots can be added, giving a better estimate of the velocity field. It is likely that the spots registered in different series of frames are spots of the same particles that have moved a short distance. Such an evaluation was performed, in the same experiment as the one shown in figure 9, after approximately 6 hour of stagnant conditions in the storage.

Figure 10a shows the resulting vector plot from three sets of velocity plots. The temperature difference to the surrounding was then about 1.1°C. Thermal stratification down to the level in the storage studied, 17 to 20 cm from the top, was very weak. The main flow is clearly directed vertically but there are some vectors with horizontal components. These could arise from real horizontal movements of the particles, but it is also possible that the horizontal vectors are caused by erroneous interpretation of the traces as discussed above, see section 2.5.

An approximation of the velocity profile in this boundary layer is achieved by plotting vertical velocity components versus distance to the wall (Fig. 10b). This result is achieved from a view of 4 cm in vertical direction. One
can see the stepwise change of velocities due to resolution in pixels, which could be improved with a more narrow view.

The best least mean square fit was found for a polynomial function of third degree, which is normally used in theory to represent the laminar velocity profile. Laminar theory (Holeman 1990) predicts the thickness of the boundary layer near an isothermal vertical surface. In water of homogenous temperature the predicted boundary layer thickness would be 8.1 mm when temperature difference between the water and the wall is 0.3°C and the characteristic length is 0.15 m. The predicted result fits well to the experimental value, roughly 8 mm as indicated in figure 10b. The same theory gives the maximum velocity 1.3 mm/s, which is higher than shown in the experiment. The discrepancy can be explained by weak stratification in the storage (Fig. 9e), which has a great impact on the boundary layer behaviour as discussed above.
Figure 10 a. The result of three different evaluations of velocities, separated in real time to give better field information. Field recorded after 3 h stand still in the same experiment as in figure 8 and 9. Particles of size 20-38 μm.

Figure 10 b. Detected velocities in the boundary layer versus distance to the wall. Least mean square fit of a polynomial of third degree added.
5. POTENTIAL FOR PIV IN THE STUDY OF HEAT STORAGE'S

The flow pattern in water heat storage's is characterised by very low velocities which change with time during charging, storing and discharging. Studies of the flow therefore require a method which can measure low velocities and there is need for a method that permits continuous recording during a complete storage cycle.

The video-based PIV-method has been found to be very suitable for studies of the flow in water heat storage's. It provides clear particle tracks, an effective tool to study the flow. On the video-tape, pictures are recorded every $1/25$ s and can be read into the computer separated in time from $0.04$ s up to several seconds. This means that very small velocities can be detected and by reading pictures from the same real time event with dissimilar time separation, different velocity levels can be detected. It is also possible to do animation's with the stored pictures, a powerful mean of showing flows that are too slow to be observed in real time.

One of the great benefits of this method is that the time intervals between video frames being analysed can be determined after the experiment. They can then be chosen to get the maximum information from the recordings. Other techniques require that the time intervals are determined during the experiment. In that case, the use of different time intervals means that the exposures must be repeated. The flow may then have changed. This is particularly true for experiments with transient flows.

Another advantage with the method used in this study is that there is no direction ambiguity. The order of the captured pictures is known and the particle spots can be given different colours for each captured time step. In this case, grey-scaling is used with particles moving from light to dark grey.

The greatest advantage with this technique is that the whole experiment can be documented for later analyses. It is possible to analyse the same real time interval many times and with different time delay between the captured frames hereby enabling detection of velocities within a wide range.

The main weakness of the method is that velocities above $0.03$ m/s can not be measured as a result of problems with particle tracing. Another weakness is the accuracy of the method mainly affected by limited resolution in cap-
tured frames, calibration of distances in the storage and the fact that the particles might not follow the flow due to different densities.

There seems to be a good potential for future development of this technique for visualisation and measurements of fluid flows. The main problem up till now has been the evaluation of velocity vectors from the data set of coordinates achieved from Image. If this could be done faster by using more powerful computers for example a SUN-workstation, this process could be run many times to give more complete velocity fields.

It could also be possible to use more than three spots of the same particle to detect a trace. In this way one would get fewer velocity vectors from each run but instead have more accurate values. For high velocities it would be useful to look at the possibilities of using high speed video recordings, or to use multiple exposures on each frame of the video film.

PIV-technique based on video recordings of particles in seeded water, is a powerful method for studies of fluid flow and velocity field in water heat storage's. The combination of particle traces and vector plots have proven to be an efficient tool in gaining knowledge about phenomena which affect the thermal behaviour.
REFERENCES


Hinsch K.D. 1993; Particle Image Velocimetry in Speckle Metrology (Ed Sirohi Rajpal S.) Ch 6 pp 235-324, Marcel Dekker, Inc.


Rasband W. 1993: NIH-Image Software Manual release 1.49, National Institute of Health Research Service Branch, NIMH,

APPENDIX A:

The PIV technique is based on the use of seeding particles in the moving fluid. The best would be if these particles perfectly matched the density of the fluid, that is were neutrally buoyant. This is very difficult to achieve since there are some other requirements to fulfil. For example the particles must scatter the light in a good way.

The particles used in these experiments are grained from a material that is offered under the name Pliolite AC and has a specified density of $\rho_p = 1.03 \cdot 10^3 \text{ kg/m}^3$. One way to achieve neutrally buoyancy would be to add for example common salt to raise the density of the fluid. This could have an impact on the piping in the circulation system and is therefore not used here.

A correct value of the density is instead determined by introducing particles in solutions of different salt concentrations. Neutrally buoyancy was achieved for the fluid density, $\rho_L = \rho_p = 1.034 \cdot 10^3 \text{ kg/m}^3$.

When the density of the fluid and the particles is known is it possible to estimate how much the particle tracks detected, differs from the movement of the fluid. These differences are due to gravitational and inertia effects. The gravitational field is conservative and has the same effect all over the storage but the inertia effects are local and due to the acceleration of the fluid, in practice when the flow direction is changed.

The first effect could be estimated by calculating the stationary falling speed of a particle in a stand still fluid. Stoke's theory is valid for spheres at low relative velocities. ($\text{Re}_d = \frac{\nu_m d \rho_L}{\mu_L} \leq 1$)

Hence the drag is given as

$$\vec{F}_D = -3 \pi d \bar{v}_f \mu_L \quad (A1)$$

the buoyancy force as

$$\vec{F}_B = \rho_L g \frac{\pi d^3}{6} \quad (A2)$$
and the weight

\[ F_s = \rho_p g \frac{\pi d^3}{6} \]  
(A3)

Force balance in vertical direction gives gravitational induced falling velocity

\[ v_f = \frac{g(\rho_p - \rho_L) d^2}{18 \mu_L} \]  
(A4)

Most of the experiments are performed with particles of size 38-76 \( \mu \)m.

(A4) implies the gravitational falling velocity 96 \( \mu \)m/s for a particle with diameter 70 \( \mu \)m in water of temperature 20°C and 24 \( \mu \)m/s for a particle with diameter 35 \( \mu \)m.

This is roughly what is found in a performed stand still experiment, with nearly negligible influence from natural convection and particles in the fraction 20-38 \( \mu \)m. Particle tracking resulted in figure A1 and A2, giving the mean value of \( v_f = 20 \) \( \mu \)m/s for the biggest particles, which is in good agreement to the theory.

Secondly if the flow is accelerated, the particles respond with an inertia delay. Assuming again validity of Stoke's law with \( g \) in equation (A4) replaced with the local acceleration \( a \) to obtain the slip velocity

\[ v_s = v_p - v_L = \frac{(\rho_p - \rho_L) a d^2}{18 \mu_L} \]  
(A5)

Force balance for a particle following the fluid in a circular orbit of radius \( r_0 \) with constant angular velocity gives

\[ r = r_0 \exp\left(\frac{\omega^2 d^2}{18 \mu} (\rho_p - \rho_L) \tau\right) \]  
(A6)

where \( \tau \) is the elapsed time.
Typical values in these experiments, where the highest velocities are found near the inlet, are

\[ \omega = \frac{\pi}{2} \text{ rad/s} \]
\[ \tau = 2s \]

giving \[ \frac{r}{r_0} = 1.0003 \]

that is a negligible deviation from the fluid trace for a particle of size 70 \( \mu \text{m} \).

Conclusion: The density difference can be neglected near the inlet and in the boundary layer near the wall, but has to be considered when the velocity field in the centre of the storage is evaluated. Here the mean fluid velocity in the experiments is of the same magnitude as the calculated falling velocity for the particles.

Figure A1. Particle tracking picture from the gradient zone during a stand still experiment.
Figure A2. The resulting velocity field from figure A1.
ABSTRACT

The main goal for this investigation was to study the possibilities of using numerical simulations for prediction of the velocity and temperature fields near the inlet of a thermally stratified water heat storage. Numerical simulations were performed with FLOW3D, a commercially available software. Calculated velocity and temperature fields were compared to the results from experiments on a model heat storage.

The comparisons between simulations and experiments show that the software can predict the main pattern of the velocity and temperature fields in the storage. This justifies the use of the software for numerical studies of phenomena that affect the mixing of hot and cold water near the inlet of the storage.
1. **INTRODUCTION**

Storing of sensible heat in water heat storage's has become fairly common in recent years and will be even more used in the future. The use of heat from solar collectors will always require some kind of heat storage and water is the most convenient storage medium to use. In district heating systems, the water heat storage has become a natural component which is used for optimising the operation from both economical and environmental point of view.

In almost every application with heat stored in water it is essential that the storage is thermally stratified to give the best performance. The degree of stratification is to a large extent determined by the mixing of hot and cold water that occurs during the initial part of a charging or discharging cycle. This is when the slope of the temperature gradient in the storage is established. The gradient is later affected by heat diffusion from hot to cold water and by natural convection caused by heat losses to the surroundings, but those effects become important only after long storage times.

The initial mixing is determined by the inertia of the incoming water and by buoyancy due the temperature difference between incoming water and the water in the storage. There is still no theoretical model that describes this mixing process, which covers the most important phenomena effecting the mixing and yet is simple enough to use on small computers in engineering work. Earlier experiments [6,7] indicate that this mixing is related to a modified Richardson number, which is the ratio between buoyancy and inertia at the inlet. With the definition of velocity and length scale that was used in [6,7] it was found that Ri-numbers smaller than about 0.25 would give strong mixing.

Application of this finding for prediction of the performance of storage's of larger size and with different inlet geometry involves difficulties with defining the characteristic length and velocity to use for calculation of the Ri-number. The problem might be studied by extensive experiments with storage's of different size and geometry. An alternative approach would be to use numerical simulation. This however is only meaningful if the numerical model has been validated through comparisons to experimental data.
The purpose of this study was to investigate the possibility of using the FLOW3D model for prediction of essential phenomena occurring during charging and discharging of a water heat storage, which had earlier been studied experimentally [11].

2. EXPERIMENTAL SET-UPS

2.1 Small model storage

Most of the experiments used for validation of the computer model were performed in a small model storage of cylindrical shape and having Plexiglas walls. The diameter of the storage was 194 mm and the height 420 mm. During charging the hot water was distributed in radial direction at the top of the storage, entering through an annular slot of diameter 97 mm and width 2 mm as shown in figure 1. The outlet was at the bottom and was also shaped as an annular slot with the same diameter but the width 4 mm. The total volume of the model storage were 12.3 litres.

The storage was run with different charging rates, from 1 storage volume in 4 h up to 1 storage volume in 1/2 h, corresponding to flow rates from 52 ml/min up to 410 ml/min. The inlet temperature was varied in the range of 25 to 50°C and the storage was normally at room temperature when the experiments started.

It was possible to vary the heat losses through the wall by circulating water of predetermined temperature on the outside of the storage. The overall heat transfer coefficient was calculated to be between 30 and 46 W/m²K depending on the temperature difference to the surrounding.

Temperatures were measured at different levels as shown in figure 2 and the data acquisition system stored the values, normally every 30 seconds.

The velocity measurements, that are fully described in [11], were performed in three steps:

1. Video recording of the particle movements in a light sheet introduced in the storage from a laser beam spread by a cylindrical lens.
2. Reading of frames from the video tape into a computer and image processing to obtain the co-ordinates of the captured particles.
3. Particle tracing to identify spots of the same particle on different frames and calculation of the velocity.

Particle trajectories can be determined by adding many frames from the video film, delayed a proper time according to the magnitude of the velocity to study. These trajectories are a powerful mean for visualising the flow as shown in figure 3 a-d.

2.2 Larger model storage

Attempts were also made to validate the computer model against experiments in a larger model storage of 1.2 m$^3$. This storage had a domed roof and the dimensions were total height 2.56 m and diameter 0.80 m. It had steel walls and was not insulated. The estimated overall heat transfer coefficient was about 10 W/m$^2$K. The inlet was designed as an annular slot with diameter 115 mm and width 17 mm and the water was entering the storage in radial direction at the top.

The inlet flow was 600 litre/hour of water at 50.2°C, giving an inlet velocity of 2.7 cm/second. The initial temperature in the storage was 21.4°C and the storage was mixed to homogeneous temperature before the experiment started.

Temperatures were measured by 49 gauges distributed from top to bottom in the storage.

Velocity measurements were made with LDA-technique. A more detailed description of this experimental set-up and the results obtained in it can be found in [7].
3. NUMERICAL SIMULATIONS

3.1 Laminar conditions

The conditions in this small storage are such that laminar flow can be expected. The governing equations to solve for this non isothermal flow are the basic equations for conservation of mass, momentum and energy, here given in index notation.

The continuity equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho U_i) = 0 \quad (1)$$

The momentum equation

$$\frac{\partial \rho U_i}{\partial t} + \frac{\partial}{\partial x_j} (\rho U_j U_i) = -B_i + \frac{\partial \sigma_{ij}}{\partial x_j} \quad (2a)$$

where $\sigma_{ij} = -\mu \delta_{ij} + \mu \left( \frac{\partial U_i}{\partial x_i} + \frac{\partial U_j}{\partial x_j} \right)$

and $B$ is the body force

The energy equation

$$\frac{\partial \rho H}{\partial t} + \frac{\partial}{\partial x_i} (\rho U_i H - \lambda \frac{\partial T}{\partial x_i}) = \frac{\partial p}{\partial t} \quad (3a)$$

where $H = h + \frac{1}{2} U_i U_i$

These represent 5 simultaneous equations for the 7 unknowns $U_i$, $p$, $T$, $\rho$, $h$.

The system is made complete by adding two algebraic equations from thermodynamics, the equation of state

$$\rho = \rho(T, p) \quad (4)$$

and the constitutive equation
\[ h = h(T,p) \]  \hspace{1cm} (5)

For the cases considered here, the fluid properties can be considered to be constant, with average values in the temperature span. The only exception is density in the buoyancy term in the momentum equation where the Boussinesq approximation is used

\[ \rho = \rho_0(1 - \beta(T - T_0)) \]  \hspace{1cm} (6)

where \( \beta \) is the coefficient of thermal expansion and \( \rho_0 \) is the density at the buoyancy reference temperature \( T_0 \).

Equations (1)-(6) is a system of non-linear, partial differential equations that in reality can only be solved with numerical techniques. For laminar flows this can be done by discretization with a reasonable amount of finite volumes.

### 3.2 Turbulent conditions

#### 3.2.1 The transport equations

For a large storage, the inlet conditions indicate turbulent behaviour of the flow. The above set of equations will still be adequate for description of the instantaneous flow field, but it is not possible with present computers to handle the details of turbulent flow.

Treatment of turbulent flows requires some kind of turbulence model. Some of the commonly used approaches will be briefly described below.

The variables in equations (1)-(6) are normally written as the sum of a mean value and a fluctuation around that value, i.e. \( \bar{U}_i = U_i + u_i \). If Reynolds averaging is applied and fluctuations in density are neglected the result will be the following set of equations

\[ \frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho U_i) = 0 \]  \hspace{1cm} (1*)
The Reynolds-averaged continuity equation is the same as the one that has not been averaged. However, the momentum and energy equations contain new so called turbulent flux terms. These are

\[ \frac{\partial \rho U_i}{\partial t} + \frac{\partial}{\partial x_j} (\rho U_j U_i) = -B_i + \frac{\partial}{\partial x_j} (\sigma_{ij} - \rho \overline{u_i u_j}) \]  

\[ \frac{\partial p H}{\partial t} + \frac{\partial}{\partial x_i} (\rho U_i H + \rho \overline{u_i h} - \lambda \frac{\partial T}{\partial x_i}) = \frac{\partial p}{\partial t} \]  

where \[ H = h + \frac{1}{2} U_i U_i + k = h + \frac{1}{2} U_i U_i + \overline{u_i u_i} \]  

Turbulence models close the equations by providing means for the computation of the Reynolds stresses and Reynolds fluxes. This is achieved with equations that describe the complex behaviour of turbulence and will always contain empirical constants.

There are two commonly used approaches

- to derive the equations for \( \overline{u_i u_j} \) and \( \overline{u_i h} \) which will contain correlation's of higher order that has to be modelled. This results in a quite general but complex model.

- to model \( \overline{u_i u_j} \) and \( \overline{u_i h} \) with other turbulence parameters, that could be described empirically or determined by semi-empirical equations, which are simpler than those for \( \overline{u_i u_j} \) and \( \overline{u_i h} \).

The turbulence models can be classified according to the number of turbulence quantities which appear as dependent variables of differential equations. The complexity of the model rises with the number of such variables.
3.2.2 The eddy viscosity concept

Many models are based on the concept of an eddy viscosity which goes back to Boussinesq [6]. The Reynolds stresses and fluxes are here modelled algebraically in terms of known mean quantities. More precise, the eddy viscosity hypothesis states that the Reynolds stresses can be linearly related to the mean velocity gradients.

\[-\rho \overline{u_i u_j} = \mu_i \left( \frac{\partial U_j}{\partial x_i} + \frac{\partial U_i}{\partial x_j} \right) - \frac{2}{3} \rho k \delta_{ij} \]  

(9)

where \( \mu_i \) is an additional viscosity, called the eddy viscosity or the turbulent viscosity and

\[ k = \frac{1}{2} \overline{u_i u_i} \text{ is the turbulent kinetic energy.} \]  

(9a)

In analogy with the eddy viscosity hypothesis, the eddy diffusivity hypothesis states that the Reynolds fluxes of a scalar are linearly related to the mean scalar gradient, for example for enthalpy

\[-\rho \overline{u_i h} = \Gamma_t \frac{\partial H}{\partial x_i} \]  

(10)

here \( \Gamma_t \) is the eddy diffusivity, usually written as

\[ \Gamma_t = \frac{\mu_i}{\sigma_t} \]  

(10a)

where \( \sigma_t \) is the turbulent Prandtl number

Subject to these hypothesis, the Reynolds averaged momentum and energy equations can be written as:

\[ \frac{\partial \rho U_i}{\partial t} + \frac{\partial}{\partial x_j} \left( \rho U_j U_i \right) = B_i - \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu_{\text{eff}} \left( \frac{\partial U_j}{\partial x_i} + \frac{\partial U_i}{\partial x_j} \right) \right) \]  

(2**)  

\[ \frac{\partial \rho H}{\partial t} + \frac{\partial}{\partial x_i} \left( \rho U_i H - \left( \frac{\lambda}{c_p} + \frac{\mu_l}{\sigma_h} \right) \frac{\partial H}{\partial x_i} \right) = \frac{\partial p}{\partial t} \]  

(3**)
where \( \mu_{\text{eff}} = \mu + \mu_t \) is the effective viscosity and \( \sigma = p + \frac{2}{3} \rho k \) a modified pressure

There are several turbulence models implemented in FLOW3D. Only the simplest of these, based on two additional transport equations were used in this study.

### 3.2.3 Models with two additional transport equations

These models are quite general but are restricted to applications where the eddy viscosity/diffusivity concept is valid (i.e. \( \mu_t \) is supposed to be isotropic and \( -\rho u_i u_j \) and \( \left( \frac{\partial U_i}{\partial x_i} + \frac{\partial U_j}{\partial x_j} \right) \) to have the same sign). These models are most tested until today and have been found useful in many engineering applications.

The k-\( \varepsilon \)-model uses the eddy-viscosity hypothesis with the assumption that \( \mu_t = C_{\mu} \rho k^2 / \varepsilon \). Two additional transport equations are solved, one for the turbulent energy \( k \) and one for the turbulence dissipation rate \( \varepsilon \). These contains terms for shear production \( P \) and production due to body forces \( G \).

The low Reynolds k-\( \varepsilon \)-model is a modification of the standard k-\( \varepsilon \)-model and introduces a damping of the eddy viscosity, a modified definition of \( \varepsilon \) which makes it approach zero at walls and modifications of source terms in the \( \varepsilon \) equation. This model covers the range \( 5000 < \text{Re} < 30000 \).

### 3.3 The software

#### 3.3.1 General description

The simulations were performed with the commercially available software FLOW3D Release 3.2 from AEA TECHNOLOGY, Harwell Laboratory UK [1]. Most of the calculations were made on a SUN Sparc IPX workstation but there was later an opportunity to run on IBM RS 6000. This decreased the computation time necessary for simulations by a factor of about 11.
The software includes a module for pre-processing (geometry and grid generation), a solution module and a module for post-processing (graphic presentation of results).

Boundary-fitted co-ordinates can be used to allow treatment of arbitrary two- and three-dimensional geometries. Cartesian or cylindrical co-ordinate systems and multi-block structuring of the physical domain is possible. The complex flow domain in physical space is mapped on to a simple (rectangular) flow domain in computational space by curvilinear co-ordinate transformations. The equations are then discretised with respect to the computational space co-ordinates.

For the discretisation of the equations above, the basis of the code is a conservative finite-volume method. All the variables are defined at the centre of the control volumes, which fills the physical domain being considered. Each equation is integrated over the control volume to obtain a discrete equation which connects the value of each variable at the centre of the control volume with the values at the centres of adjacent control volumes.

The software permits fully 3-dimensional simulations but in this case it has been possible to use 2-dimensional simulations due to the cylindrical symmetry. This leads to more reasonable computation times. Since the main interest was focused on simulating the initial mixing near the inlet, it has not been necessary to model the whole storage in all the simulations. In many simulations only the volume from the top down to the centre of the storage has been modelled and a pressure boundary used at that level.

There is a very useful option of adding own FORTRAN routines in a set of modules provided and designed for special purposes. These FORTRAN modules gives access to the variables in the program and enables the calculation of user defined variables that are not included in a normal run.

3.3.2 Application of the model to thermal storage's

In all the simulations described below, the default options in the program were normally used, see APPENDIX C. For example, the hybrid differencing is always used for discretisation of the convection terms and backward differencing used for discretisation in time. There are a number of other
options available and some of them are tested in this investigation but no improvement was found in using them. One exception is that in these simulations the program was allowed to do up to 60 iterations in each time step to reach convergence, default is 30. When other options were used that are not default it will be stated when the simulation is described.

3.3.3 General experiences

When this investigation started there was only access to Release 3.0 of the FLOW3D program, which worked in single precision. This version gave large problems when applied on this problem with large density gradients. The problems were first detected as strange wavy velocity fields, which were not recognised from the experiments. This appeared in the vicinity of the gradient zone in the storage and could also be detected as strange variations in the pressure field. Although the total mass flows across the boundary of cells were within the stipulated mass residuals and indicated perfect mass continuity in the storage, the mass flow passing cross sections at different levels in the storage calculated using nodal velocities at the centre of each cell could differ from the actual mass flow by a factor of 10.

This problem was found to be related to the velocity-pressure coupling algorithm and the interpolation method used to obtain velocity components at control volume faces from those in control volume centres. The problem is known as Rhie-Chow errors and can almost be eliminated with the improved Rhie-Chow interpolation method implemented in release 3.2. Still there are discrepancies of as much as 10% in some calculations, between the actual mass flow and that calculated from nodal velocities.

When the calculations were extended to a larger storage with the same number of grid points as in the small one the same problems were detected. The only way found to solve the problem was to use finer grids on the expense of long computer time.
4. COMPARISONS BETWEEN EXPERIMENTS AND SIMULATIONS

4.1 Charging experiments

Simulations were performed for four charging experiments in the small storage. The predicted temperatures and velocities were compared to results from experiments. Three of the experiments were made for quite stable charging conditions, that is high temperature difference between incoming water and the water in the storage and a low flow rate. The fourth experiment was performed with the opposite charging conditions, low temperature difference and a comparably high charging rate, giving more mixing of hot and cold water near the inlet.

One simulation was performed on a larger storage for stable charging conditions. The predicted temperature evolution was compared to the results of the experiment.

4.1.1 Stable charging

The first comparison was made in an experiment with the following charging conditions:

- Initial temperature in the storage: 20.6°C
- Inlet temperature: 40.0°C
- Temperature difference to the surrounding: ≈ 0°C
- Inlet flow: 180 ml/min
- Inlet velocity: 5 mm/s

The mass flow corresponds to a charging cycle of about 70 minutes. Figure 3a-d shows particle traces achieved at four different times during the initial part of the charging. The seeding particles were introduced with the hot water, which can be seen in figure 3a where particles are present only in the warm stream at the roof.

The simulations were performed with the assumption of laminar, incompressible, buoyant and transient flow. Two-dimensional calculations with cylindrical co-ordinates was applied. The whole storage was modelled with three blocks and with a total of 2880 control volumes in the computa-
tion domain. A pressure boundary was set at the outlet. Zero heat losses were assumed and the initial conditions were homogenous temperature and zero velocity in the whole storage. The inlet was defined to have the same velocity over the whole cross section and both velocity and temperature were kept constant during the simulation.

Small time steps of 0.2 seconds were used in the beginning of the simulation when the buoyancy effect was strongest. Convergence to total mass residuals smaller than $1.0 \times 10^{-8}$ kg/s was achieved in about 700 s CPU time for 1 s of real time advance with the program run on a SUN-IPX workstation. (The inlet flow was $3.0 \times 10^{-3}$ kg/s). After 10 s of real time advance the time step was raised to 0.5 s and convergence was achieved in about 200s CPU-time for each time step, that is 400 s CPU-time/1 s real time.

Figure 4 a-d show the predicted velocity field and Figure 5 a-d the predicted temperature field at the same real time as in the experiment above.

The main flow pattern predicted in the simulations is clearly the same as in the experiment during this initial part of a charging cycle with strong buoyancy compared to inertia. Strong buoyancy forces prevent the first incoming water from penetrating down into the cold water and a narrow stream of hot water flows on top of the cold water (Fig. 3a, 4a). When this stream reaches the wall of the storage it is redirected downwards. Since the inlet flow is radially distributed the velocity will be quite low due to continuity, when the water reaches the wall. The buoyancy forces will therefore redirect the stream towards the centre of the storage.

Again due to continuity, if this stream towards the centre is not widened the velocity would have to increase. However inertia will counteract this effect and tend to widen the stream. The stream is redirected towards the wall again by the inlet plate, as can be seen in figure 3b and 4b.

When more hot water is fed to the storage, the buoyancy forces acting on the incoming water will decrease and the incoming stream becomes wider. (Fig. 3c and 4c). A boundary layer is established at the roof and tends to direct the stream of incoming water a bit more downwards.

Figure 5a-d show the predicted temperature fields at the same instants as the velocity fields shown in figure 4 a-d. Figure 4a shows a big convection cell
typical for a back step. This cell could cause a lot of mixing but as figure 5a shows there is no hot water going downwards, mixing into cold water. The mixing that occurs takes place when some cold water is entering the warm stream (re-entrainment) and also at the front of the warm stream.

Later on, the widening of the incoming stream appears in the hot part of the storage and does not cause mixing. In fact all down-ward streams are effectively stopped by the gradient zone, which now is established and is moving downwards in the storage with the mean velocity of the charging.

Figure 3d and 4d show that the simulations can not predict all the different velocity fields that can arise in the hot part of the storage, where no buoyancy forces are present. The simulations show very smooth velocity fields due to constant boundary conditions throughout the simulation while in the experiment these boundary conditions can vary. Inlet flow and temperature as well as heat losses through the wall can undergo small variations and cause changes in the field, which are very difficult to predict. However, these differences in the velocity field are observed in the hot part of the storage where it does not effect the mixing of hot and cold water to some noticeable extent.

Figure 6 shows the velocity field documented with video based Particle Image Velocimetry technique after 4 minutes of charging and figure 7a the corresponding velocity and figure 7b the corresponding temperature field predicted with FLOW3D. The resulting velocity field from the experiments is represented by unevenly distributed velocity vectors due to the random distribution of seeding particles.

The predicted velocities are of the same magnitude as the measured velocities and the overall movement close to the inlet is quite similar, although the simulations did not predict the weak upward redirection of the flow in front of the inlet. Further down the stream towards the centre of the storage below the inlet plate seems to be overestimated by the simulation program.

In the lower part of the figures there is a flow towards the wall which seems to be much stronger in the experiment than in simulations. This is probably due to small heat losses through the wall that after all are present in the experiment. This would cause a down going flow in a boundary layer at the wall below the gradient zone. Water had to be fed to this flow in the lower
part of the gradient zone as shown in figure 6. The program was run with
the assumption of zero heat losses and could therefore not predict this flow.

Comparisons for this experiment, with a large temperature difference be­
tween the incoming water and the water in the storage and low inlet veloc­
ities, indicate that the overall behaviour of the flow can be predicted in a
good way. The resemblance is however not perfect but this could merely be
explained by the fact that the true boundary conditions in the experiment
are very difficult to document and thereby to reproduce in the simulations.

The usefulness of the software for predicting the thermal behaviour of
water heat storage's can be judged by examining the temperature evolution
for the gauges in the storage and compare them to the simulated evolution.
For this purpose it was necessary to use the option of adding own
FORTRAN routines to the simulation program. There are a lot of prepared
FORTRAN modules suitable for different tasks, in this case the USRTRN
routine was used which is called after each time step and gives access to the
variables needed. The temperatures in the control volumes that represent
the locations of the temperature gauges was captured by the routine at
predetermined real time intervals and stored for further evaluation.

Figure 8 shows the temperature evolution detected in the experiment
above. Temperatures for 5 gauges in the storage are shown, as they were
measured in the experiment and predicted by FLOW3D. The flow measure­
ments during the experiment gave probably too high values since the tem­
perature rise for the gauges is a bit delayed in the experiment. FLOW3D
seems to predict less mixing near the inlet than shown in the experiment
(steeper temperature gradient). These differences will be more discussed in
the next comparison.

The next comparison of the temperature evolution was made in an experi­
ment performed with the aimed charging rate 180 ml/min, meaning that
the storage would be completely charged in about 70 minutes. A control
based on the experimental temperature evolution for gauges in the storage
showed that the real charging rate was slightly less than 160 ml/min, as a
mean value for the whole charging cycle. The conditions for the simulation
were:
### Initial Conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial temperature in the storage:</td>
<td>24.6 °C</td>
</tr>
<tr>
<td>Inlet temperature:</td>
<td>42.9 °C</td>
</tr>
<tr>
<td>Temperature difference to the surrounding:</td>
<td>3.0 °C</td>
</tr>
<tr>
<td>Inlet flow:</td>
<td>160 ml/min</td>
</tr>
<tr>
<td>Inlet velocity:</td>
<td>4.4 mm/s</td>
</tr>
<tr>
<td>Estimated heat losses through the wall:</td>
<td>150 W/m²</td>
</tr>
<tr>
<td>Estimated heat losses through the roof:</td>
<td>10 W/m²</td>
</tr>
</tbody>
</table>

The water on the outside of the storage was fed at a rate giving the same mean velocity on the outside as on the inside of the storage. The initial temperature of the water on the outside of the storage was about 21.6°C and the water fed to the outside had a temperature of 40.0°C. In this way it was possible to maintain almost constant temperature difference over the whole height of the storage while the zone between hot and cold water was moving downwards.

The simulations were performed with the assumption of laminar, incompressible, buoyant and transient flow. Two-dimensional calculations with cylindrical co-ordinates was applied. The storage was modelled down to the centre level, 0.21 m from the top, where a pressure boundary was set. 1440 cells were used to resolve the computational domain.

Initial conditions in the storage were set to zero velocity and 24.6°C initial temperature in the whole domain. The perimeter was defined as a wall with a constant heat flux of 150 W/m² out of the storage and the roof as a wall with the heat flux 10 W/m². The inlet conditions were first simulated with constant temperature from time zero and constant velocity all over the cross section of the inlet. This resulted in a predicted temperature stratification that was stronger than the measured stratification. This could however be explained by the fact that the inlet temperature did not rise to its maximum value in zero time in the experiment due to mixing of hot and cold water in the tubing before the inlet.

This effect was considered in later simulations, shown below, by use of the optional module for specification of boundary conditions, USRBCS. The inlet temperature was varied in the same way as the inlet temperature in the experiment. The results from experiments and simulations are compared in figure 9.
The figure shows the time variation of temperatures at different levels in the storage. When the temperature starts to rise it means that the lower part of the gradient zone reaches this gauge. The temperature of the gauge then increases until the gradient zone has passed this level and the gauge is surrounded by hot water. The figure also shows that the temperature for gauges further down in the storage does not reach the inlet temperature, indicating a weak stratification above the gradient zone.

There is good similarity between measured and predicted evolution of temperatures indicating that the simulation program predicts the mixing of hot and cold water that occurs near the inlet in a good way. The weak stratification that appears above the gradient zone is also predicted in the simulations although it is weaker than the measured stratification.

However, there are some differences which deserve attention. First there seems to be a certain time delay between the rising of temperatures in the experiment and the simulation. This can be explained by the fact that it was not possible to locate the centres of the computation cells exactly at the same level as the corresponding gauge in the storage and that the real position of the gauges was determined with some inaccuracy. Second, the weaker stratification above the gradient zone shown in the simulations is to some extent due to the heat that was stored in the Plexiglas wall in the experiment and this effect was not modelled in the simulations, although it should be possible.

Comparisons to some of the old measurements in the larger storage described in [7] was also used for validation of the software. Results from some recent simulations are shown in appendix A. Here it is concluded that extension of the calculations to larger stratified storage's can not be readily done, because of problems with Rhie-Chow errors mentioned in section 3.3.3.

If the software is used for simulations on large stratified storage's, the results must be carefully examined and continuity at cell centres checked. The solutions can be converged with small mass residuals and continuity fulfilled at cell faces and still the solution might contain large errors.
4.1.2 *Extreme charging with mixing*

One experiment performed in the small storage was designed to get the most unstable charging conditions that was possible to achieve within the limits set by the experimental equipment.

The experimental conditions were:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial temperature in the storage</td>
<td>23.8 °C</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>25.1 °C</td>
</tr>
<tr>
<td>Temperature of the surrounding water</td>
<td>23.0 °C</td>
</tr>
<tr>
<td>Room temperature</td>
<td>22.0 °C</td>
</tr>
<tr>
<td>Inlet flow</td>
<td>400 ml/min</td>
</tr>
<tr>
<td>Inlet velocity</td>
<td>10.7 mm/s</td>
</tr>
<tr>
<td>Overall heat transfer coefficient for the wall</td>
<td>50 W/m²K</td>
</tr>
<tr>
<td>Overall heat transfer coefficient the roof</td>
<td>0.9 W/m²K</td>
</tr>
</tbody>
</table>

The inlet flow corresponded to a charging cycle of half an hour. The surrounding water volume was maintained at an approximately constant temperature.

Figure 10 shows the measured and the predicted temperature evolution for 5 gauges in the storage. The figure shows that the storage was slightly stratified before the experiment started. This was due to heat losses during the stand still period that preceded the experiment.

The overall assumptions for the simulations were the same as those stated above for the small storage and the storage was modelled down to the centre using 1440 cells and a pressure boundary was set at that level. The perimeter of the storage was defined as a wall with the overall heat transfer coefficient $U=50$ W/m²K and loosing heat to a surrounding at 23.0°C. The roof was given the overall heat transfer coefficient 0.9 W/m²K and was loosing heat to room air at 22.0°C.

Figure 11 a and b show the velocity field as it was detected with the Particle Image Velocimetry technique after 6 respectively 12 minutes of charging. Figure 12 a and b show the corresponding velocity field predicted with FLOW3D and figure 12 c and d the predicted temperature field.
There is a good similarity between the measured and the predicted velocity fields at both times. The vertical extension of the convection cell, which has the greatest impact on the mixing of hot and cold water is essentially the same. The horizontal extension of the cell is however greater in the simulations than the extension found in the measurements.

The predicted temperature distributions show that the vertical extension of the convection cell in both cases is limited within the warm volume of the storage, that is above the gradient zone. When this zone moves downwards the convection cell can increase its vertical extension. If no buoyancy is present the cell will stop growing when the kinetic energy in the incoming flow is consumed by viscous dissipation.

4.2 Discharging experiment

One discharging experiment, that was performed in the small storage, was also simulated. The experimental conditions were:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial temperature in the storage:</td>
<td>38.0 °C</td>
</tr>
<tr>
<td>Inlet temperature:</td>
<td>24.6 °C</td>
</tr>
<tr>
<td>Temperature difference to the surrounding:</td>
<td>1.6 °C</td>
</tr>
<tr>
<td>Inlet flow:</td>
<td>180 ml/min</td>
</tr>
<tr>
<td>Inlet velocity:</td>
<td>1.8 mm/s</td>
</tr>
<tr>
<td>Estimated heat losses through the wall:</td>
<td>50 W/m²</td>
</tr>
</tbody>
</table>

The simulations were performed with the assumptions above and half of the storage modelled using 1440 cells. The measured velocity field is shown in figure 13 and the one predicted with FLOW3D is shown in figure 14, both after 14 minutes of discharging of the storage. The main flow field is well predicted by FLOW3D in both magnitude and direction of velocities.

The principal difference between charging and discharging experiments in the interaction between the boundary layer at the wall and the incoming flow is also correctly predicted with FLOW3D. The flow in the boundary layer near the wall will always be directed downwards in real storage's since they are loosing heat to the surrounding. The boundary layer flow thus interacts with the incoming flow in charging experiments and the incoming flow will feed water into the boundary layer flow.
During discharging experiments these both flows will counteract since they are going in opposite directions when they meet at the bottom of the storage. In the beginning of a discharging cycle the boundary layer flow will be weak and the two flows will build up a stream going up quite close to the wall. The weak boundary layer flow is due to a small height for the layer to grow. This height is limited by the gradient zone and the bottom of the storage.

As the gradient zone moves upwards the boundary layer flow will grow and in this small storage become dominant compared to the incoming flow. In the figures shown the boundary layer has become dominant and the resulting flow is directed upwards immediately in front of the inlet. This type of interaction between the two flows will give more mixing of hot and cold water, than the mixing that occurs at the inlet during charging experiments under equivalent conditions.

5. NUMERICAL STUDY OF VARIABLES THAT AFFECT THE INITIAL MIXING

These simulations are described in APPENDIX B and were performed for the same model storage that was used for the validation of the software.

The comparisons between different simulations in this investigation were based on an exergy efficiency defined as the actual exergy content in the storage divided by the highest possible content if no mixing had occurred, that is if all heat in the storage would have inlet temperature.

The parameters studied that were related to the flow were: inlet temperature, inlet velocity, inlet mass flow and temperature difference to the water in the storage and those related to the geometry were: inlet diameter, width of the inlet slot and the diameter of the storage.

It was concluded that in stable chargings of this small storage there were detectable differences between the different simulations in the initial phase of the charging process. In most of the simulations these differences were evened during the continued charging and most of the simulations showed the same exergy efficiency after charging the storage to a bit less than 25%. This was concluded to be an effect of the heat diffusion from hot to cold
water which tended to compensate for the differences that were achieved in the initial phase.

There were however two simulations that gave smaller exergy efficiencies than the other. One was simulating a storage charged with very low flow from an inlet with small diameter. The other simulated a storage with large storage diameter but normal inlet diameter and normal flow.

Both these cases causes the first incoming jet to be spread over a large horizontal area when it enters the storage. This will cause more mixing in the contact between hot and cold water and strong heat diffusion since the temperature gradient becomes large in this contact region. There is also a larger distance up to the wall in these cases and there will be more mixing due to the widening of the jet and due to reintrainment of colder water into the jet. These differences will also be evened by heat diffusion but only if the heat is stored for a long time in the storage.

New scales for velocity and length were proposed for use in the calculations of the Richardson number. The characteristic length was proposed to be the width of the inlet slot and the characteristic velocity to be essentially the velocity of the jet from the inlet when it reaches the wall of the storage.

It was concluded that the Ri-number should only be used for assessment of the possibility that the charging conditions would cause strong mixing caused by redirection of the inlet jet downwards along the wall into the cold part of the storage. On basis of simple theoretical reasoning, the stability limit would be at $R_i = 0.5$ or lower. The results of the simulations indicate that such unstable charging is obtained at least for $R_i$ less than 0.05. The practical importance of the initial mixing obtained for $R_i = 0.05$ is not necessarily large since the exergy efficiency will recover to almost the same levels as for stable charging. The simulation for $R_i = 0.003$ however resulted in severe mixing and a lasting significant reduction of the exergy efficiency.

The transient temperature fields generated by the simulation illustrate the differences in behaviour of the storage's for stable and unstable charging conditions. They also show how stratification is established after some initial mixing at $R_i = 0.05$. 
6. **CONCLUSIONS**

This investigation shows that FLOW3D is suitable for calculations of temperature and velocity fields near the inlet of the small model storage used in the PIV experiments. It is therefore possible to use the program for investigations of the different parameters that affect the mixing of hot and cold water during the initial phase of a charging or discharging cycle.

For this small storage the degree of turbulence is so low that the main flow field and the temperature distribution can be well predicted with calculations using the laminar option.

The extension of the simulations to larger storage's is more doubtful due to problems with small grid induced velocities. These will arise when grids of moderate refinement, here less than about 5,000 computation cells, are used and strong density gradients are present. Strong density gradients are always present in the gradient zone in stratified storage's and the predicted temperature distribution will indicate too weak stratification if extremely fine grids are not used. Before simulations with this software are performed for larger storage's, more work has to be put into solving this problem and finding a cheaper way than grid refinement to master these errors. If the software is used for simulations on large stratified storage's, the results must be carefully examined and continuity at cell centres has to be checked.

An exergy efficiency was introduced and as a measure of the performance of the storage with respect to maintaining the temperature level of the hot water charged into the storage.

Simulations performed for the small model storage show that the differences in exergy efficiency in the initial phase of the charging caused by different charging conditions were much reduced during the continued charging cycle. This was concluded to be an effect of heat diffusion. The same effect can be expected in larger storage's even if the process there would be slower.

It was found that the Richardson number was not suitable to use for predictions of the initial mixing for stable charging conditions. However it proved to be useful to separate stable and unstable charging conditions.
New scales for the velocity and the length to use in the calculations of the Richardson number were proposed. These scales are more related to the conditions at the inlet. The new scales are considered to be useful also for large storage's. The limit for the Ri-number where charging starts to be unstable can be expected to be in the range of 0.5-0.05. Lower values of Ri indicates unstable charging conditions. Even at Ri 0.05 however stratification was quickly obtained after some initial mixing in the small storage studied in the simulations. When Ri was further reduced to 0.003, severe mixing was found in the simulation.

The stability limit should be further investigated for larger storage's by further simulations and experiments.

ACKNOWLEDGEMENT

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REFERENCES


Figure 1. Design of the inlet.

Figure 2. Positions for the temperature gauges in the set-up.
Figure 3a. Particle traces achieved after 20 seconds of charging. (1 frame/second).

Figure 3b. Particle traces achieved after one minute of charging. (1 frame/2 seconds).
Figure 3c. Particle traces achieved after 3 minutes 40 seconds of charging. (1 frame/second).

Figure 3d. Particle traces achieved after 15 minutes of charging. (1 frame/second).
Figure 4a. Velocity field predicted by FLOW3d. Corresponds to the measured field in fig 3a.

Figure 4b. Velocity field predicted by FLOW3d. Corresponds to the measured field in fig 3b.
Figure 4c. Velocity field predicted by FLOW3d. Corresponds to the measured field in fig 3c.

Figure 4d. Velocity field predicted by FLOW3d. Corresponds to the measured field in fig 3d.
Figure 5a. Temperature field predicted by FLOW3d. Calculated for the same real time as the velocity fields in figure 3a and 4a.

Figure 5b. Temperature field predicted by FLOW3d. Calculated for the same real time as the velocity fields in figure 3b and 4b.
Figure 5c. Temperature field predicted by FLOW3d. Calculated for the same real time as the velocity fields in figure 3c and 4c.

Figure 5d. Temperature field predicted by FLOW3d. Calculated for the same real time as the velocity fields in figure 3d and 4d.
Figure 6. Velocity field detected with PIV-technique after four minutes of charging in an experiment with inlet velocity 5 mm/s, inlet temperature 40 °C and initial temperature in the storage 21.6 °C. A vector of length 1 mm in the scale of the figure correspond to the velocity 1 mm/s in the upper part of the picture (down to 8 mm) and 0.5 mm/s in the lower part.
Figure 7. Velocity filed predicted with FLOW3d. Compare to the field achieved in the experiment, shown in figure 6.
Figure 8. Temperature evolution for 5 gauges in the storage, measured respectively predicted with FLOW3d. Inlet flow 180 ml/min.

Figure 9. Temperature evolution for five gauges in the storage as they are recorded in the experiment and predicted by FLOW3d. Charging rate was 160 ml/min giving a mean velocity in the range of 5.4 m/min. Temperature limiting is 70 °C.
Figure 10. Temperature evolution for 5 gauges in the storage, measured respectively predicted with FLOW3d. Inlet flow 400 ml/min.
Figure 11a. Velocity field detected with PIV-technique after 6 minutes of charging in an experiment with inlet velocity 11 mm/s, inlet temperature 25.1 °C and initial temperature in the storage was 23.8 °C. A vector of length 1 mm in the scale of the figure correspond to the velocity 0.5 mm/s.
Figure 11b. Velocity field detected with PIV-technique after 12 minutes of charging in the experiment shown in fig 11a.
Figure 12a. Velocity field predicted with FLOW3d at the same real time as in figure 11a. The vector scale is the same as in figure 11a.
Figure 12b. Velocity field predicted with FLOW3d at the same real time as in figure 11b. The vector scale is the same as in figure 11b.
Figure 12c. Temperature field predicted with FLOW3d at the same real time as in figure 11a and 12a.

Figure 12d. Temperature field predicted with FLOW3d at the same real time as in figure 11b and 12b.
Figure 13. Velocity field measured with PIV-technique in the discharging experiment.

Figure 14. Velocity field predicted with FLOW3d in the discharging experiment.
APPENDIX A

SIMULATIONS OF THE TEMPERATURE EVOLUTION IN THE LARGER STORAGE

The program was also validated against experiments performed in the larger storage of 1.2 m³. Up till now only a few such simulations have been made. An example is presented below and compared to the experimental results for the temperature evolution's at three temperature gauges in the storage.

The computational domain was resolved by 2490 cells, with the finest grid close to the inlet. The domain was divided in two blocks, one from the top down to 0.89 m and the other from this level down to the bottom. 1500 cells were used in the first block with contracting subdivision in radial direction (30 cells) and expanding in axial direction (50 cells). This configuration was chosen in order to have at least some cells in the boundary layer at the wall and some cells to define the inlet. It is not possible in the program to define boundary conditions on parts of a cell. Since the top wall of the first block was not plane this meant that the cells in this block were skewed as shown in figure A1.

The second block was divided in the same way in radial direction which is necessary to be able to "glue" block 1 and block 2 together. In axial direction uniform sub-division using 33 cells with the vertical extension 5 cm was used.

Simulations were performed with the laminar option and with the turbulent option using the "Low Reynolds number turbulence model". The laminar simulations were run to simulate 92 real time minutes of charging which demanded approximately 34 hours of CPU-time on the RS 6000 system. The time steps varied between 0.1 s in the beginning and 1.0 s at the end of the run. The turbulent calculations were only run to simulate 36 minutes of charging using 29 hours of CPU-time on the same system. Inlet Reynolds number was not higher than 1650 but in radial jets, transition to turbulence can occur at even lower Re numbers.

The results are shown in figure A2. There was practically no difference between the results for laminar and turbulent flow. The simulation predicts a
much wider gradient zone than found in the experiments. This means that more time is needed for the gradient zone to pass each gauge.

The results were not very encouraging since the aim was to use the software for further simulation. An examination of the mass continuity calculated with values at the centre of the cells showed larger discrepancies in mass flow than was accepted in the simulations for the small storage (at most 10%). The differences were up to 100% in the gradient zone for the laminar simulation and up to 30% for turbulent case. Nevertheless the predicted temperature evolution was practically the same in the laminar and the turbulent case.

The velocity field also showed some wavy variations in the velocities that were not detected in the experiments. These small grid induced velocities will lead to mixing of hot and cold water in the gradient zone which results in a widening of the zone in the simulations.

The problems were similar to those encountered in the simulation with release 3.0 of the program for the small storage. Then the errors were due to calculations in single precision (which made it impossible to achieve acceptably small mass residuals) and to the interpolation method used for computation of the velocity component on control volume faces from those in control volume centres. The latter problem was encountered only when strong density gradients were present, for example in stratified water heat storage's. Both these problems were solved in release 3.2 which allowed use of double precision and an improved interpolation method.

Four possible explanations for the problems found in the simulations for the larger storage have been considered.

1. The interpolation method is sensitive to the skewness of the grid and the used grid could be the reason for the problem.
2. The time steps used could be too large even if the solutions seemed to be converged.
3. The differencing scheme used is of too low order.
4. The calculation grid used is too coarse, especially in the vertical direction where the strongest gradients normally are found. The maximum vertical extension of cells in these calculations was ten times larger, 5 cm in this storage compared to 5 mm in the small storage. This could perhaps
give steps in density between cells that are too large for the interpolation method.

If the problem is of the first type this would be simple to handle for most storage's. In order to study the effect of eliminating the skew grid a new simulation was performed, this time on a storage with the same volume but with the doomed top replaced by a volume of cylindrical shape with a plane roof. The largest vertical extension of the cells was here 2 cm and the storage was modelled down to one meter below the top to save computation time. 1350 cells were used in the computation domain and layers of cells were parallel to the density stratification. The result of this simulation deviated even more from the experimental results.

The second explanation is less likely since the tests of release 3.0 indicated that these errors were hardly affected by a decrease of the time step. This is probably true also for the larger storage. No attempt to use smaller time steps has been made however.

The third possible explanation was studied by running a simulation with use of a higher order differencing scheme for the discretisation of the convective-advective terms. The result with the 3rd order 'QUICK' scheme showed a larger discrepancy in mass continuity and did not solve the problem.

If the fourth explanation is valid it would be severe for the possibilities to use this software for simulations on large stratified storage's. Using a grid of the same spatial resolution as in the small storage would lead to very long computation times which is undesirable for economical reasons.

An attempt was nevertheless made to use a refined grid. These calculations were performed for a grid with vertical extension of cells 0.4 mm or smaller. The same domain as in the second simulation was used which required more than 10000 computation cells. The simulations were continued for 24 minutes of real time by use of more than 50 hours of CPU-time on the RS 6000 system.

The result from these simulations are shown in figure A3 together with the experimental results. The discrepancy for mass continuity calculated at cell centres was here less than 1%. The temperature is quite well predicted and
the simulations indicate that it is possible to do simulations on this size of a storage with acceptable results. The computational effort is so large that it is doubtful if the software should be used for such simulation in the present form, if no other solution than refinement of the grid is possible.

The software is however continuously developed and a solution to the problem might therefore be implemented in the near future. It might also be possible to find a solution within the present software. This problem will be addressed in further work.

The capacity of computers is however fast increasing and the problem may be solved in this way in a near future.

Figure A1. The calculation grid used for the larger storage, here shown from the top down to 0.95 m below the top.
Figure A2. Temperature evolution for three gauges in the storage. Results from measurements and from simulations.

Figure A3. Results with finer grid.
APPENDIX B

NUMERICAL STUDY OF VARIABLES THAT AFFECT THE INITIAL MIXING

Introduction

The good resemblance that was found between experimental data and simulations for the small storage justify the use of the software for simulations on variables that were not possible to study in the experiments.

The initial goal was to extend the simulations to larger storage's but the experiences from simulations at larger storage's does not encourage this step. Simulations are possible to do for the medium sized model storage of 1.2 m$^3$ but with long times needed for the calculations.

Simulations for full size storage's of >500 m$^3$ has to wait until the problems with maintaining mass continuity in the stratified region are solved. This could be done either by amendments of the software to allow use of grids with reasonable refinement or with the use of computers of larger capacity.

Nevertheless it should be possible to use simulation for studies of the adequacy of the Richardson number for modelling of the initial mixing in the storage and about the scales that should be used in that case. It should also be possible to investigate the importance of this initial mixing on the degree of stratification in the storage.

The Richardson number, $R_i$

The Ri-number applied on heat storage's is often given [14] as

$$R_i = \frac{g \cdot \beta \cdot \Delta T \cdot h}{v^2}$$

The distance between the inlet and the outlet has been proposed for the length scale, $h$. The inlet velocity is used for the velocity scale, $v$, and temperature difference between incoming water and the water in the storage for temperature scale, $\Delta T$. A high value for the Richardson number is often
associated with high stratification as a result of little mixing between hot and cold water, see [7,14]. The Richardson number is also used for assessment of the stability of the charging, see [7,14]. The Ri-number could also be written as \( Ri = \frac{\Delta \rho \cdot g \cdot h}{\rho v^2} \) and interpreted as essentially the quotient between the work against the buoyancy forces and the kinetic energy of the water when it starts the vertical penetration into the water of higher density. This indicates that the hot water would be able to penetrate downwards a distance \( h \) for a value of Ri of 0.5 if no mixing between hot water and cold water should occur. With mixing, a lower value for the Richardson number is needed to give the same penetration. The critical value for the Ri-number, below which unstable conditions appear at the inlet, is often said to be about 0.25.

This formulation of the Ri-number implies that the mixing at the inlet is not dependent on the mass flow, the inlet slot or the inlet diameter as long as the inlet velocity is constant. Neither would the diameter of the storage affect this mixing. Experimental studies where these parameters are varied will be time consuming and expensive. Numerical simulation can not eliminate the need for experimental work but can be useful at least as a basis for planning of experiments by highlighting important phenomena.

**Measures for the degree of stratification**

For quantitative comparisons between simulations of different cases, an aggregate measure for the degree of stratification in the storage is useful.

The measures discussed in [7], were developed for use when only temperature measurements were available. The mixing volume, penetration depth, thickness of the gradient zone and the temperature gradient had to be calculated based on a quite coarse resolution of the temperature field. The gauges were normally staggered only in one vertical plane.

These measures of the stratification then had to be determined when each gauge could be expected to represent the temperature in one horizontal plane, that is at a level where the gradient zone was established. In this way it was not possible to follow the mixing in the very beginning of the charging cycle and the experiments could only be compared when the
gradient zone was established. It was possible to relate the mixing to the Ri-number when the charging caused heavy mixing but not to separate experiments with stable charging. This was not satisfactory but seemed to be the only possibility with the information that was available.

The simulations however give the temperature of every computation cell which can be used to value the temperature level for the heat from the start of the charging. It should thus be possible to study cases with stable chargings, which of course should be the normal case, to see the impact of the important parameters.

As suggested in [7] it was chosen to use the Carnot factor to value the heat in each cell. This value was normalised through division with the maximum possible Carnot factor, that is the value achieved if the heat in the cell had been at inlet temperature. The ratio so obtained can be considered as an exergy efficiency, since it compares the actual exergy content in the storage to the maximum possible value achieved if all heat had the inlet temperature. This normalisation made it possible to compare simulations with different charging conditions.

The exergy efficiency $\eta_{\text{ex}}$ was calculated as

$$\eta_{\text{ex}} = \frac{\sum_{\text{cells}} Q_{\text{cell}} \cdot (1 - T_{\text{ref}} / T_{\text{cell}})}{\sum_{\text{cells}} Q_{\text{cell}} \cdot (1 - T_{\text{ref}} / T_{\text{inlet}})}$$

where

- $Q_{\text{cell}} = \text{actual energy content in each cell} = H_{\text{cell}}$
- $T_{\text{ref}} = \text{reference temperature for exergy and enthalpy} = \text{initial temperature in the storage}$
- $T_{\text{inlet}} = \text{inlet temperature} = \text{maximum possible temperature}$
Simulations

All the simulations were made for a storage with a volume of 0.012 m$^3$, i.e. the same as for the small model storage used in the experiments described in [11]. The time evolution of the exergy efficiency was used for comparison of the results of the simulations.

The efficiency is given related to a normalised time achieved by dividing the real time with the time needed for filling of the storage. When the mass flows were the same in the compared simulations this meant that the real times for the different simulations also were the same. The calculations were continued until the storage’s were filled to about 25% at which stage the influence from the inlet on the gradient zone was considered to be negligible for stable charging.

The simulations were performed with the assumption of zero heat losses. The reference temperature was set to the initial temperature in the storage both in the calculation of enthalpy and in the calculation of exergy. These calculations were performed with predetermined real time intervals by use of the optional FORTRAN routine USRTRN every 2 seconds in the beginning of the run, later every 5, 10 and 20 second.

The effect of the following variations of the most important variables were studied by the simulation

1. The mass flow was varied by change of the inlet slot or the inlet diameter while the inlet velocity and other variables were kept constant.
2. The inlet velocity was varied by change of the inlet slot or the inlet diameter while mass flow and other variables were kept constant.
3. The temperature difference between incoming water and the water in the storage was varied while all the other variables were constant.
4. The diameter of the storage was varied while all the other variables were kept constant.

The simulations were performed with the same computational grid as was used in the validation of the program.
The initial temperature in the whole storage was set to 24.4°C. Maximum inlet temperature was set from time zero of the charging. The chosen values for the charging flow were centred around the most often used flow in the previous simulations, 180 ml/min. This corresponds to about one hour charging time for the model storage of volume 12.3 litres. The reference case and the simulated range for variables are given in the table below.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Reference case</th>
<th>Range simulated</th>
</tr>
</thead>
<tbody>
<tr>
<td>Charging volume flow</td>
<td>180 ml/min</td>
<td>18 - 675 ml/min</td>
</tr>
<tr>
<td>Inlet velocity</td>
<td>4.85 mm/s</td>
<td>1.29 - 100 mm/s</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>42.9 °C</td>
<td>25.1 - 74.6 °C</td>
</tr>
<tr>
<td>Inlet slot width</td>
<td>2 mm</td>
<td>0.92 - 7.5 mm</td>
</tr>
<tr>
<td>Inlet diameter</td>
<td>98 mm</td>
<td>10 - 98 mm</td>
</tr>
<tr>
<td>Storage diameter</td>
<td>194 mm</td>
<td>194 - 274 mm</td>
</tr>
<tr>
<td>Ri-number*</td>
<td>58</td>
<td>0.13 - 15 000</td>
</tr>
</tbody>
</table>

*Ri-number with length scale = storage height = 0.42 m
velocity scale = inlet velocity

Most of the simulations were made for conditions where stable charging was expected, i.e. for high values of the Richardson number. Two simulations were made for conditions where the low value of the Richardson number indicated unstable charging.

**Effect of the mass flow**

Since the inlet velocity was considered to have great impact on the mixing near the inlet it was kept constant during these simulations. The mass flow was varied by changing the inlet area, either by changing the inlet slot or the inlet diameter.

Figure 1 shows the results from five such simulations with different mass flows. The time scale used has been normalised related to the flow so that the storage is filled to the same degree at equal normalised times. The simulations show that the exergy efficiency drops quickly but soon recovers and will slowly increase to just below 90% at normalised time 0.25.
For perfect charging, the exergy efficiency would have been maintained constant at 1.0 from the start of the charging since no mixing would occur and thus no degradation of temperature for the heat. The figure shows that the least mixing in the initial phase of the charging will be obtained for the reference case.

The loss in exergy shown by the simulation is the combined effect of mixing and heat diffusion. The temperature gradients are large when the hot incoming water meets the cold water in the storage which gives strong diffusion. When the charging rate is low, the mixing of hot and cold water is expected to be weak but the diffusion will anyhow decrease the exergy of the heat in the storage. This is clearly shown for the smallest flow were the lowest value for the exergy was predicted. After this initial phase the exergy in the storage will start to increase when the exergy flow into the storage is larger than the exergy loss due to mixing and diffusion.

When the simulations were continued up to a charging of the storage of about 25%, the exergy efficiency's became the same. The interpretation for this is that the diffusion tends to even out differences achieved during the initial phase of the charging. If little mixing is achieved it results in large temperature gradients and thus strong heat diffusion which tends to even out the differences. Due to long computation time, the simulation for the smallest flow with the small inlet diameter was stopped when the storage was charged to 8% only. The tendency to recover exergy efficiency is similar with that shown by the other simulations.

For the smallest flows it will take a long time to fill the storage to a certain percentage and the diffusion will act during a long time. On the other hand, if a fast charging is followed by a long stand still period the exergy loss by diffusion and the result will be quite the same, if the total time that the gradient zone stays in the storage are equal.

It is also interesting to compare the two simulations that had roughly the same flow but achieved in different ways. The initial exergy losses were roughly the same but the continuations were somewhat different. The flow of 87 ml/min was achieved by decrease of the diameter of the inlet slot and the flow of 85 ml/min by decrease of the width of this slot. It is likely that the lower value of the exergy efficiency for the case with the smaller diameter was due to the fact that the horizontal area of the storage at the inlet
level was larger in this case. This would give more mixing and heat diffusion and result in a lower exergy for the heat in the storage. The distance from the inlet to the wall was larger in this case and the spreading of the radial jet acted on a longer distance, which would give more mixing.

The geometry and the other conditions for these simulations were within the range validated against experiments. The results shown are therefore expected to be true, at least for this small storage.

The conclusion would be that for stable charging of this small storage, the diffusion is mainly responsible for the resulting exergy content and thereby the temperature distribution in the storage when more than about 10% of the storage has been charged. It is also evident that the Richardson number as defined in [7,14] is not sufficient for modelling of the charging transient. The simulations gave different results for the initial phase of the charging even if all the variables in the Ri-number as defined in [7,14] were constant.

**Effect of the inlet velocity**

The different inlet velocities were achieved by change of the inlet diameter and the width of the inlet slot, while the mass flow was kept constant.

Figure 2 shows the results from these simulations. The lowest value of the exergy was achieved for the lowest inlet velocity. To get this low velocity the width of the inlet slot was increased which gave a large frontal area of the incoming flow for mixing and diffusion. The fact that the exergy starts rising at an earlier state in this case could be interpreted as if there was less mixing in this case. A late increase of the exergy efficiency indicates more mixing.

With this interpretation of the curves, the highest inlet velocity gave most mixing. This is also what to expect since it gives the lowest value for the Ri-number.

Even if the different inlet velocities gave quite different initial values for the exergy efficiency the resulting values after charging the storage to 25% were the same. This is in agreement with the simulations discussed earlier.
Again bearing in mind that these are results for a small storage they imply that differences in the initial phase will be evened out by diffusion during the continued charging. This conclusion is valid for inlet conditions that give stable charging.

**Effect of the inlet temperature**

These simulations were performed under so called normal conditions except that the inlet temperatures were varied.

The results are shown in figure 3 and 4. Figure 4 shows a close up view of the interesting part in figure 3. The figures show that the different simulations gave the same exergy efficiency after a short time.

The close up view shows that the evolution's differ during the initial phase. The lowest exergy efficiency was achieved for the highest temperature. This contradicts the expectation that a high Ri-number through a high temperature difference, gives better stratification. It appears that the larger temperature difference leads to more exergy loss through diffusion.

A low charging temperature gives a more rapid recovery of the exergy efficiency. This may be explained by stronger mixing during the initial phase of the charging which reduces the temperature gradient and thereby the further exergy loss by diffusion.

**Effect of the storage diameter**

Two values of the storage diameter were used in the simulations, the reference diameter and a diameter giving the double cross section area of the storage. All other variables were kept constant.

Figure 5 and 6 show the resulting evolutions of the exergy efficiency. The close up view in figure 6 shows that the increase of the exergy efficiency starts much later for the larger diameter which agrees with the statement above. The doubled area means that the radial jet reaches the wall later in the case with larger storage diameter. The jet would be expected to reach the
wall at about twice the time achieved for the lower diameter. The spreading of the jet on the way up to the wall might delay it a bit.

The larger horizontal area for the large diameter gives bigger contact area between the radial jet and the cold water and therefore larger exergy losses by mixing and diffusion. This gives lower exergy efficiency from the beginning of the charging which will be maintained until the storage is completely charged.

**Effect of charging for unstable conditions**

Two simulation were performed to test whether strong mixing can be predicted. The reference geometry was used, the inlet velocity was set to 100 mm/s and two different temperature differences to the water in the storage were studied. The inlet flow was much larger in this case so that the storage was charged to 25% in 46 seconds.

An examination of the diagrams for the simulations show that with this storage it should be possible to reach well above 85% for the exergy efficiency when the storage is charged to about 25%.

The resulting evolutions of the exergy efficiency are shown in figure 7. The simulated conditions were:

<table>
<thead>
<tr>
<th></th>
<th>Simulation 1</th>
<th>Simulation 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet velocity</td>
<td>100 mm/s</td>
<td>100 mm/s</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>1.3 °C</td>
<td>18.5 °C</td>
</tr>
<tr>
<td>Ri-number</td>
<td>0.13</td>
<td>2.4</td>
</tr>
</tbody>
</table>

With a critical Ri-number of 0.25, the first simulation would give unstable charging whereas the second would be stable.

Figure 7 shows that the higher temperature difference gave a more stable charging which could be expected. The curve for the temperature difference 18.5°C show that these conditions does not cause severe mixing in the simulations. The exergy efficiency at the end of the simulated period is compar-
able or even better than those achieved for very small flow and inlet diameter, see figure 1, and for the large storage diameter, see figure 5.

Discussion about the Ri-number

The simulations show that the Ri-number in the form suggested in [7,12] is not suitable for predictions of the result of the initial phase in stable charging. There are more variables that must be considered, for example the diameter of the inlet slot and the diameter of the storage. The width of the inlet slot will also have some effect on the resulting temperature distribution.

The Ri-number might still be useful for assessment of the stability of the charging. Stable charging means little forced mixing during the initial phase.

Even for that purpose the choice of variables in the Ri-number suggested in [14] is doubtful. The simulations presented here show that for example the diameter of the storage has great impact on the result. Alternative definitions for the length and velocity scale to use in the Ri-number can be considered however. The discussion is limited to storage’s of cylindrical shape.

The choice of the distance between inlets for the length h means that unstable charging in principle should mix the whole storage. It is possible to set a much more narrow limit for the mixing.

The simulations for stable charging show that different initial responses of the storage tend to be evened out during the continued charging. The critical time is when the jet reaches the wall and is redirected downwards. If the inertia then is too high the redirected jet will penetrate downwards before it is horizontally redirected again by buoyancy forces. The downward flow determines the initial mixing.

The minimum penetration is obtained if the direction of the flow is immediately changed to a horizontal movement towards the centre. This behaviour is observed in the flow field measurements and predicted by FLOW3d in stable chargings. This can be used to define the appropriate length scale in the Ri-number. The jet must be allowed to penetrate down-
wards a distance corresponding to the width of the jet when it reaches the wall, not more.

The jet is widened towards the wall by spreading of the jet and by reintrainment of water into the jet. This widening is found to be about 11% of the travelled radial distance even for turbulent jets. Such results were found for jets in an isothermal surrounding, see [5]. In the small model storage and with normal inlet velocities this would give a widening of the jet from 2 mm at the inlet up to about 7 mm at the wall if there were no buoyancy forces.

The widening is however decreased by buoyancy forces which is observed in the experiments and predicted by FLOW3d. For wide inlet flows and small velocities the width can even decrease close to the inlet. For stable chargings the width of the jet when it reaches the wall is of the same order of magnitude as the inlet slot. This should be valid even for larger storages of cylindrical shape even if so called floating inlets are used. In that case the widening is affected by buoyancy forces from hot water above and cold water below the inlet.

On basis of this it is suggested to use the width of the inlet slot for the length scale \( h \) in calculations of the Ri-number.

In storage’s of cylindrical shape the velocity will be lower when the water reaches the wall than it was at the inlet. It appears as more reasonable to use the jet velocity at this wall as a measure of the inertia of the inlet flow than to use the actual inlet velocity. It is therefore suggested that the jet velocity at the wall is used for calculation of the critical Richardson number. It is possible to estimate this velocity considering the continuity and the widening of the jet. However buoyancy will reduce the spreading during this initial phase and no correction is suggested for this widening of the jet.

The velocity to use in the Richardson number can then be calculated as:

\[
v = \frac{d_{\text{inl}}}{D_{\text{sto}}} v_{\text{inl}}
\]

where \( d_{\text{inl}} \) is the inlet diameter and \( D_{\text{sto}} \) the storage diameter.
There seem to be no reason to change the temperature scale to use in the Richardson number since the buoyancy is essentially determined by the temperature difference between incoming water and the water in the storage when the warm jet starts penetrating downwards at the wall.

When the Ri-numbers are calculated for the stable experiments simulated above it results in Ri-numbers between 1 and 288 with the new scales and between 6 and 15 000 with the old ones. That is, both ways to calculate the Ri-number indicate stable charging if the critical value is 0.25.

The simulated transient temperature field for stable charging can be illustrated by figure 8a-d which show the development during the first two minutes.

The pattern is quite different for the two simulations with lower Ri-numbers. With the new definition, the Ri-numbers for these cases are 0.003 and 0.05 compared to 0.13 and 2.4 with the definition suggested in [14]. Figures 9a-d show the transient up to 25% charging for the lowest Ri-number and 10a-d that for the next lowest. The dramatic penetration of the hot water into the cold part of the storage is evident for the lowest Ri-number. For the next lowest Ri-number, the jet penetrates into the cold water but the buoyancy forces stabilise the charging later in the transient. At 25% charging, figure 10d, the storage is reasonably stratified. Apparently, the critical Ri-number is above 0.05 with the new definition of the scales but the practical importance of the instability obtained at this Ri-number is not large since the exergy efficiency recovers to the same level as for stable charging (compare results shown in figures 5 and 7).

It is interesting to compare the temperature transients for the large storage diameter where the exergy efficiency is shown in figure 5 with the simulation for the bigger temperature difference with exergy efficiency shown in figure 7. The result at the end of the simulated period is quite the same but the reasons for the lower exergy efficiency’s compared to other simulations are not the same. This can be illustrated by the predicted temperature fields shown in figure 8 and figure 10. The same volume is fed to the storage’s in both figures.

Figure 10, illustrating the case with high inlet velocity show that the water starts penetrating downwards when the jet reaches the wall, while no such
tendency is shown in figure 8. The reason for the low exergy efficiency when
the inlet velocity is high is obviously forced mixing. When the inlet veloc-
ity is low and the storage diameter large it is likely that the heat diffusion is
responsible for much of the exergy losses.

CONCLUSIONS

The simulations above were performed on a small storage of cylindrical
shape with an inlet formed as an annular slot, placed at the top of the
storage. The extension of the results to larger storage's and storage's of other
shapes has to be performed very carefully and the general validity of the
results should not be taken for granted.

The behaviour of real storage's in service is merely determined by the tem-
perature distribution inside the storage and at the end the behaviour should
be judged in economical terms. The exergy efficiency is a suitable measure
for the value of maintaining a high temperature for the heat. Other
measures could have been used, for example the mean temperature for the
heat stored, but the results would have been the same.

Heat diffusion seems to have a greater impact on the thermal behaviour
during the initial phase of the charging than was suggested in earlier work,
but this is probably to a large extent an effect of the scale for the simulations.
In larger storage's with wider inlets the exergy losses due to heat diffusion
across the horizontal contact area will represent a much smaller fraction of
the incoming exergy flow. More work is needed for the scaling of this effect
to larger storage's or for simulation of this effect in a larger scale.

The simulations show that as long as the charging conditions are stable, as
judged by the Ri-number, there are no large differences between the separate
charging conditions after 10-25% charging of the storage. The initial differ-
ences disappear to a large extent during the continued charging.

The effect should be the same in larger storage's even if the process is slower
due to the wider gradient zone. The temperature difference between the hot
and the cold water is normally higher and the residence time for the water
in the storage is longer in a full size storage. The conclusion would thus be
that if the whole storage cycle is considered the heat diffusion will eliminate much of the initial differences achieved if the charging conditions are stable.

Even if the simulations show that unstable charging conditions is the most important reason for a low exergy efficiency, it is still important to consider other factors which tend to reduce the exergy efficiency.

It is not for instance advisable to try to achieve very low inlet velocities by widening the inlet slot. A wider slot gives a wider contact area between hot and cold water during the initial phase.

Also, the inlet diameter should not be too small since a small inlet diameter gives more widening of the incoming jet and a bigger horizontal contact area between the first incoming hot water and the cold water. More work is suggested on this matter to find out if it is possible to give some recommendations for the choice of inlet diameter related to the storage diameter.

The Richardson number is not suitable for predictions of the exergy efficiency for stable chargings, since it does not account for all important variables. This is a conclusion that should be valid also for larger storage's.

The new scales proposed for the length and the velocity in the Ri-number are more related to the geometry where the mixing occurs and should enable more precise predictions of the risk for undesirable mixing of hot and cold water during the initial phase of the charging. These scales are suitable to use also for larger storage's.

For the small cylindrical storage simulated here, the critical value for the Ri-number, calculated with the new scales, is above 0.05. At this Ri-number the hot water does not penetrate far into the cold water, however. As charging progresses the storage becomes stratified and the exergy efficiency reaches about the same levels as for stable charging. For Ri-numbers that are an order of magnitude lower, the mixing becomes severe and the exergy efficiencies much lower.

In larger storage's the incoming jet will be turbulent and more work is needed for determination of the critical Ri-number. It is necessary to evaluate the influence from the widening of the jet and from reintrainment for
the turbulent case. The main flow towards the wall is expected to be roughly the same however.

Taking into consideration the weak influence of the Ri-number on the exergy efficiency when some penetration of hot water occurs at the wall, it is expected that a storage designed for a Ri-number of 0.05-0.5 will give reasonable exergy efficiencies also for the turbulent case. Additional work is needed to verify this.

![Graph](image)

**Figure 1.** Results from simulations with FLOW3D. The inlet velocity was kept constant and the flow 87 ml/min and 18 ml/min were achieved by changing the inlet diameter and the other two flows by changing the width of the inlet slot. The normal flow was 180 ml/min, inlet slot 2 mm and inlet diameter 97 mm, which gives the inlet velocity 4.85 mm/s.
Figure 2. Simulations of varying velocities with constant mass flow. Velocities 1.29, 4.85 and 10.5 mm/s were achieved by change of the width of the inlet slot and 10.0 and 48.1 by change of its diameter, from the reference values.

Figure 3. Simulations with different inlet temperatures. The other variables were kept constant.
Figure 4. Close up view of the graph shown in figure 3.

Figure 5. Simulations with two different storage diameters. Other conditions were from the reference case.
Figure 6. Close up view of the above figure.

Figure 7. Exergy evolution for two charging with extreme inlet velocity and different temperature difference to the water in the storage. Richardson numbers calculated with the old and the new definition (in parenthesis) are given.
Figure 8a. Predicted temperature field after 20 seconds charging of a storage with reference geometry except for storage diameter $D = 274$ mm. Inlet flow was 180 ml/min giving the inlet velocity 4.85 mm/s. $Ri = 40$. Compare to figure 9a and 10a.
Figure 8b. The same experiment as in figure 8a after 1 minute of charging. Compare to figure 9b and 10b.
Figure 8c. The same experiment as in figure 8a after 2 minutes of charging. Compare to figure 9c and 10c.
Figure 8d. The same experiment as in figure 8a after 16 minutes of charging. Compare to figure 9d and 10d.
Figure 9a. Predicted temperature field after 0.95s seconds charging of a storage with reference geometry. Inlet velocity was 100 mm/s giving the inlet flow 3708 ml/min. Ri = 0.003. Compare to figure 8a and 10a.
Figure 9b. The same experiment as in figure 9a after 2.9 seconds of charging. Compare to figure 8b. and 10b
Figure 9c. The same experiment as in figure 9a after 5.8 seconds of charging. Compare to figure 8c and 10c.
Figure 9d. The same experiment as in figure 9a after 46 seconds of charging. Compare to figure 8d and 10d.
Figure 10a. Predicted temperature field after 0.95s seconds charging of a storage with reference geometry. Inlet velocity was 100 mm/s giving the inlet flow 3708 ml/min. Ri = 0.05. Compare to figure 8a and 9a.
Figure 10b. The same experiment as in figure 10a after 2.9 seconds of charging. Compare to figure 8b and 9b.
Figure 10c. The same experiment as in figure 10a after 5.8 seconds of charging. Compare to figure 8c and 9c.
Figure 10d. The same experiment as in figure 10a after 46 seconds of charging. Compare to figure 8d and 9d.
APPENDIX C

Some used default options in the software.

Differencing scheme:  'HYBRID'  all equations except for density
          'UPWIND'  density

Equation solvers:  'STONE'  U,V and W velocity, enthalpy
          'ICCG'  pressure
          'LINE SOLVER'  turbulence quantities

Number of sweeps for the inner iterations

MAXIMUM NUMBER  5  all eq. except for pressure
MAXIMUM NUMBER  30  pressure

Pressure correction:  'SIMPLEC'

Reduction factors:  0.25  all equations except for
                    0.1  pressure and enthalpy

Rhie-Chow switches:  'IMPROVED'
                    'QUADRATIC EXTRAPOLATION'

Convergence testing on mass residuals.

FIXED TIME STEPPING and BACKWARD DIFFERENCE discretisation in time

Default values were used for TURBULENCE CONSTANTS (Except  C3 =1.0
instead of 0.0), TURBULENT PRANDTL NUMBER and UNDER
RELAXATION FACTORS.